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DESIGN · PRODUCTION · MATERIALS

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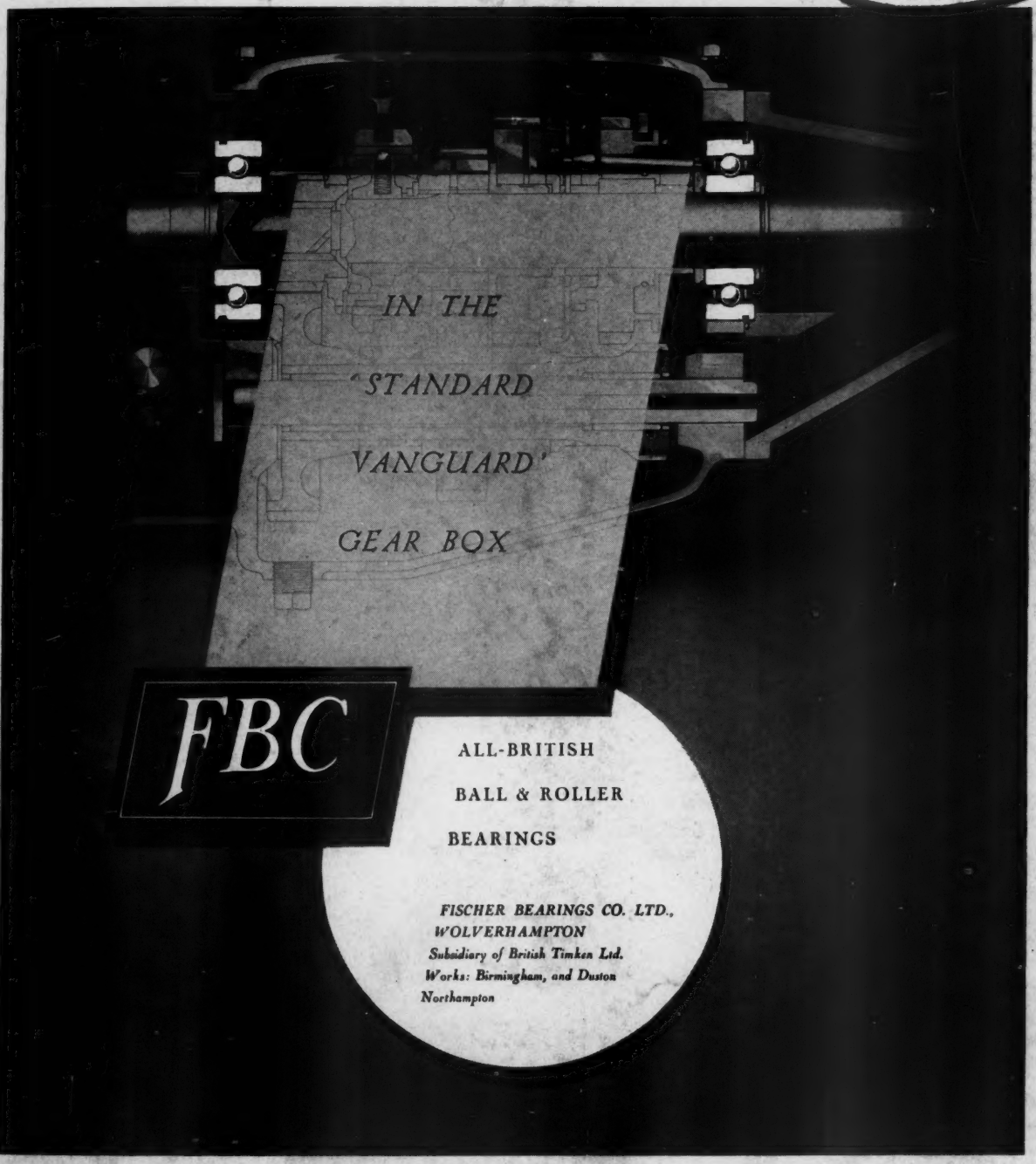

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AUTOMOBILE ENGINEER

Design, Materials, Production Methods, and Works Equipment

Consulting Editor: W. L. FISHER, M.I.Mech.E., M.S.A.E.

Editorial Staff: R. T. CARR, M.Sc., A.M.Inst.C.E., A.M.I.Mech.E., J. B. DUNCAN, T. K. GARRETT, A.M.I.Mech.E., A.F.R.Ae.S., F. C. SHEFFIELD

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The Grease Gun

THE Show period is inevitably a time of technical stock-taking, and in general terms the record of automobile engineers the world over is a satisfactory one. Throughout the civilized world, and even beyond, road vehicles provide an essential service that grows more valuable day by day. Reliability standards are higher than ever, essential durability, excluding superficial finishes, is excellent, and from any fair and reasonable viewpoint the motor vehicle must be regarded as an extremely satisfactory product. There remain however certain problems that are still by no means fully solved and perhaps the most obtrusive is the lubrication of steering and front suspension mechanisms. Here, all the circumstances work against the designer in any attempt to provide a full solution, and this is doubtless the reason why so little progress has been made.

Excluding for the moment automatic and semi-automatic oiling systems, things are very much as they were a quarter of a century ago, with the further drawback that in the meantime the number of points demanding attention has in most cases been increased. The matter is, however, of considerable importance to the user, for a variety of reasons.

With the majority of cars, the grease gun and nipple constitute the usual provision, but this is a system fraught with many serious disabilities. For one thing, when a car is delivered, the only certainty about such installations is that they will be found to be more or less inoperative. Usually the nipples, and possibly also the holes in the respective parts, will be painted over and blocked. Very often, the operative edges of the nipples themselves will be damaged. Nothing short of a nipple-by-nipple inspection, plus conscientious servicing by the owner will put the system into working condition. When this is accomplished, a combination of strength, agility and suppleness is demanded each time the grease gun is applied. To these admirable but by no means general, gifts must be added a complete indifference to every kind of filth, plus a disregard for personal safety in the matter of odd knocks, grazes, and so forth.

The alternative to this demoralizing performance is to entrust the work to a service station, but the organizations that can safely be relied upon to do the job properly are relatively rare. In most cases, the car is returned with the nipple localities buried in grease but this, alas, cannot be taken as satisfactory evidence. There is still no guarantee that any lubricant has penetrated to the working surfaces. Nipples that are not easily visible, or are somewhat inaccessible, will probably have received no attention. In any event, the surplus grease will fall on the garage floor, or on the rims and tyres, whence it will be thrown about on the underside of the vehicle generally.

In case these grim facts should invite correspondence from the sponsors of automatic and semi-automatic oiling systems, it should be mentioned that even when satisfactory in operation, these do not entirely solve the problem. For one thing, appreciable extra cost is entailed, secondly it is doubtful whether oil, even of heavy-duty quality and in reasonable and regular supply, is quite the most suitable type of lubricant for this particular work. It has to be borne in mind that, in the main, the parts of steering and suspension mechanisms do not fall into the category of rotating bearings. There is a measure of rotation, admittedly, but much of the work of these bearing surfaces consists of sustaining repeated and heavy impact blows on one specific area of the mating surfaces, in short, sheer localized hammering.

WALTER L. FISHER, M.I.Mech.E., M.S.A.E.
Editor *"Automobile Engineer"* 1914-1951

After many years in the Editorial Chair, the retirement of W. L. Fisher is announced as from October 1951. From that date Mr. Fisher will be available to "The Automobile Engineer" in a consultative capacity.

Mr. Fisher was educated at Merchant Taylors School and served a three year indentured premium apprenticeship, with two years following later at City and Guilds Technical College. He subsequently held appointments at D. Napier and Son, The Standard Motor Co., The Daimler Co., Alfred Herbert Ltd., Ryknield Bus Co., and others. He became Editor of the "Automobile Engineer" in 1914, a position he has held continuously until his retirement this month. Mr. Fisher's talents are not to be lost to automobile engineering. He has decided to set up in practice as a consultant on vehicle design and allied subjects.

Conceivably there may be a solution in the provision of one or two small, high-pressure chambers integral with the parts involved, cutting out all pipe work. These chambers, which should be readily and accessibly charged through a screw cap, might employ the same cap to supply pressure by means of a spanner. In basic principle, they would constitute a kind of super edition of the old-fashioned grease cup. They would, however, hold a far larger charge of grease and be so designed that very high pressures could be applied readily and without any spectacular contortion or muscular development being demanded of the operator. About four of such charging chambers per vehicle would deal with the steering and front suspension.

As regards the more hidden and almost completely neglected grease points underneath the vehicle, there would appear to be no ready solution to this problem from the owner's point of view. It will be up to the designer and component supplier to do away with the need for routine greasing. The needle roller universal, charged for a lifetime, would meet the demands of the universal joints of the propeller shaft. The sliding spline can be dealt with by designing it into the rear end of the gearbox or its extension.

Limited Mileage

A further point in connection with the grease gun system generally, is that even after undergoing the pains and penalties inseparable from conscientious greasing, only a limited mileage of fully satisfactory driving is provided. After about 300 miles the whole lugubrious performance must be repeated if pleasurable steering is to be maintained. A hidden chapter, one upon which it would be kinder not to dwell, is that dealing with the charging of the gun. This always depressing operation has been rendered even more sticky and hazardous by the introduction of high-duty grease. Use of this glutinous and tenacious compound ensures greatly increased life to swivel pins, steering joints and wishbone bearings. Unfortunately, it entails a correspondingly shortened life for the man behind the gun.

Having jacked up the vehicle, covered the head, face, and ears with a protective wrap, placed mats, rugs, old overcoats or sacks on the floor, the unhappy owner is ready to essay the first thrust. Alas, the gun proves to be charged with a mixture of grease and air, and no grease is emitted. Once again, the victim disentangles himself, dismantles the gun, and sets about the task of endeavouring to fill the gun with grease only. Sufficient practice, plus a few cunningly shaped pieces of wood, will together result in the evolution of a technique of satisfactory gun filling.

The point is, however, in view of the many inducements to neglect, how many owners have the pertinacity to ensure that the requisite attention is given to this job of greasing. In practice, the answer is very few, and undue wear on swivels, steering linkage, suspensions and, ultimately, front tyres are the result. The fact that only by full and conscientious attention to greasing, the steering pivots in particular, is pleasant, satisfactory handling secured, is appreciated only by the enthusiast. For the ordinary motorist, in nine cases out of ten, the job is only half done at best, and at worst, simply neglected.

Mechanical Handling

FOR many years the handling of material as it passed along a detail machining line received much less attention than the handling to effect transfer from one part of a factory to another. This is no longer true of certain organizations in the automobile industry where the handling of material throughout a complex of machining processes has in recent years been studied in great detail. Developments such as those carried out by Ford Motor Co. Ltd. for the production of Zephyr and Consul engines and by Vauxhall Motors Ltd. for detail machining on components for both passenger car and Bedford truck engines show that there is now great awareness of the importance of handling material during the machining sequence.

Transfer machining has, of course, played a great part in bringing this about. The transfer machine must necessarily pass the work mechanically from one station to the next. It must also be arranged to effect accurate location and secure clamping automatically. Since the work is passed mechanically from station to station, purely longitudinal movement is practically essential, since movement in any other direction would create additional problems. As a result "tunnel" type work fixtures must be used so that the work can move into position from one end and then, after the machining, pass out at the other end. Since this type of work fixture has been so successful on transfer machines, it has been a logical development to employ such fixtures on single-station machines.

In terms of capital expenditure, transfer machines are probably only justified economically in the very large organizations. There is, however, no reason why small organizations should not apply some of the lessons that have been learned. The one that lends itself most readily to adoption is the use of open-ended work fixtures. Experience shows that fixtures of this type are almost invariably much more open than the box and bridge type fixtures that are loaded at right angles to the machine line. They therefore reduce loading and unloading times, an important saving where the actual machining times are low as they most generally are in the automobile industry. In an appreciable number of applications the time saving on these factors may quite well be sufficient to allow one operator to tend two machines, instead of one as previously.

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THE TRIUMPH MAYFLOWER

A 1¼-litre Chassis of Unitary Construction

AFTER the war the manufacturers of the Standard Triumph cars produced the Triumph 1800 and Roadster. They then concentrated all their production capacity on the Renown and Vanguard cars, each with a 2,088 c.c. engine. Such a high degree of standardization naturally led to considerable economy in production. However, in view of the economic developments in both the national and international sphere during the post-war period, it was decided that attention should be given to the possibility of entering the market with a car of the rather more popular 1¼ litre class. There were already several models in this category well established in the public favour, and it was logical, therefore, to produce a car that was different. The knife-edge style of body was accordingly chosen, coupled with that extra degree of refinement associated with the name of Triumph.

SPECIFICATION

ENGINE: Four cylinders. Bore and stroke 63.0× 100.0 mm. Swept volume 1,247 c.c. Maximum b.h.p. 38 at 4,200 r.p.m. Maximum b.m.e.p. and torque 116 lb/in and 710 lb/in at 2,200 r.p.m. Compression ratio 6.8:1. Three-bearing crankshaft. Side valves. Solex downdraught carburettor.

CLUTCH: Borg and Beck, single dry plate, 7¼ in diameter.

GEARBOX: Three forward synchromesh speeds and normal reverse. Ratios: top 5.125, second 8.56, first 18.14, reverse 21.06.

REAR AXLE: Hypoid bevel.

SUSPENSION: Coil-and-wishbone independent front, semi-elliptic rear. Girling telescopic shock absorbers DAS 3/30 front and DAS 7/33 rear.

STEERING: Cam and follower.

BRAKES: 8 in diameter by 1½ in wide Lockheed Hydraulic, 2 L.S. at front.

TYRES: Dunlop 5.50 by 15.00.

DIMENSIONS: Wheelbase 7 ft 0 in. Track, front 3 ft 10 in, rear 4 ft 1 in. Weight of complete car 2,016 lb., dry.

Engine

The four-cylinder, side valve engine has a bore and stroke of 63.0× 100.0 mm, giving a swept volume of 1,247 c.c. The maximum torque is 710 lb/in at 2,200 r.p.m. corresponding to a maximum b.m.e.p. of 116 lb/sq in. The engine develops its peak power of 38 b.h.p. at 4,200 r.p.m. with a compression ratio of 6.8 : 1.

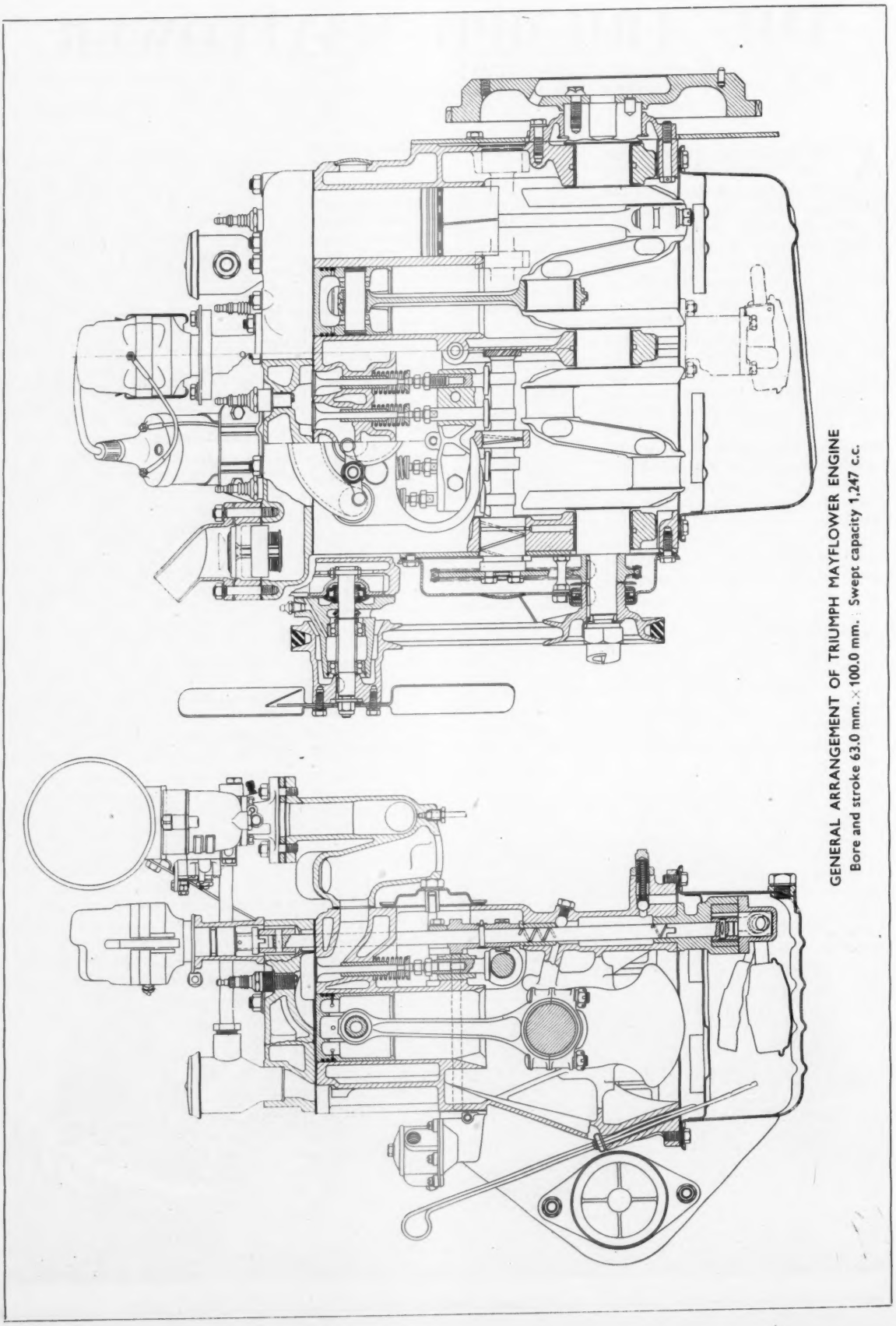
An integral cylinder block and crankcase casting, of chromium iron, extends 2¾ in below the centre line of the crankshaft. The water jackets completely encircle the block and pass between the second and third cylinders. Cast holes align with similar holes in the head for the flow of cooling water.

A fabricated, sheet-steel oil filler is pressed into the head, the oil flowing to the sump through a vertical passage cast in the cylinder head and block adjacent to No. 3 and No. 4 cylinders. The bores of the cylinders are machined in the conventional manner and hone finished. The minimum wall thickness is 0.21 in.

The front face of the crankcase is machined to provide, in conjunction with the usual joint washer, front plate and timing cover, an oil-tight joint. This face is completed at its lower end by a bolted-on, zinc diecast bridge piece which must, of course, be removed to dismantle the front journal bearing assembly and to withdraw the crankshaft. Each end of the bridge piece has cast-in slots into which hard-wood wedges are driven for sealing purposes. Shellac jointing compound is applied to the faces of the diecasting, and to the wedges, to prevent oil leaks.

At the rear end, the crankcase is closed by a small, bolted-on aluminium casting. This arrangement minimizes the machining at the face joint. The casting also incorporates the scroll-

Triumph Mayflower Saloon.



GENERAL ARRANGEMENT OF TRIUMPH MAYFLOWER ENGINE
Bore and stroke 63.0 mm. × 100.0 mm. Swept capacity 1,247 c.c.

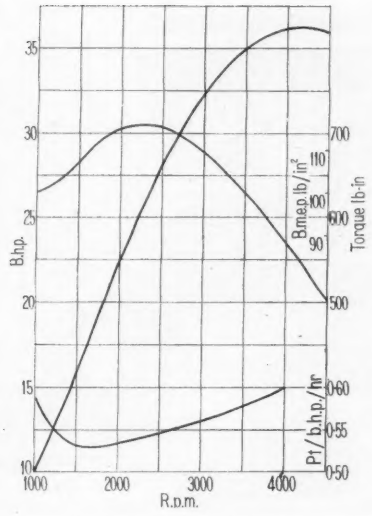
type, crankshaft rear seal, and a lip around the inner end of its bore prevents oil from draining over the rotating sealing surfaces. A die-cast bridge piece is used in the same manner as at the front end to complete the lower joint. The sandwich plate bolted to the rear end of the crankcase serves only to support and enclose the front end of the clutch bell-housing, recesses cast in the crankcase reducing the area required to be machined for this purpose.

Bosses in the crankcase form the camshaft bearings, the rearmost being fitted with a plug to prevent oil from passing into the clutch housing. The crankcase walls are well ribbed to support the three main bearings, and the cylinder skirts are recessed to clear the connecting rod. The crankshaft is offset $\frac{1}{4}$ in to the side remote from the valve gear. Vandervell steel-backed, white-metal bearing shells are held in position by caps with bolts and spring washers. Projections on the top and bottom half-shells engage in slots machined in the caps and bearing supports respectively.

Integral balance weights are employed; the En 16 crankshaft having three main journals. The principal dimensions are: crank pin diameter $1\frac{3}{4}$ in, big-end bearing length $\frac{15}{16}$ in, main journal diameter 2 in, centre main bearing length $1\frac{1}{4}$ in, end main bearing lengths $1\frac{3}{8}$ in. Four bolts locked by tab washers secure the fly-wheel which is located by a dowel. The starter teeth are machined on the periphery of the fly-wheel and induction hardened. Provision is made for the fly-wheel to be turned through 90 deg when the engine is overhauled.

Machined to $2\frac{7}{8}$ in diameter, the rear end of the crankshaft has the usual oil-return scroll and an integral oil thrower ring. The spigot hole for the front end of the clutch has an Oilite bearing.

End location of the crankshaft is by semi-circular, steel-backed, white-metal thrust washers mounted each side of the rear journal bearing cap. This arrangement simplifies the fitting of the crankshaft, connecting rod, and piston assembly. The front



Engine performance curves.

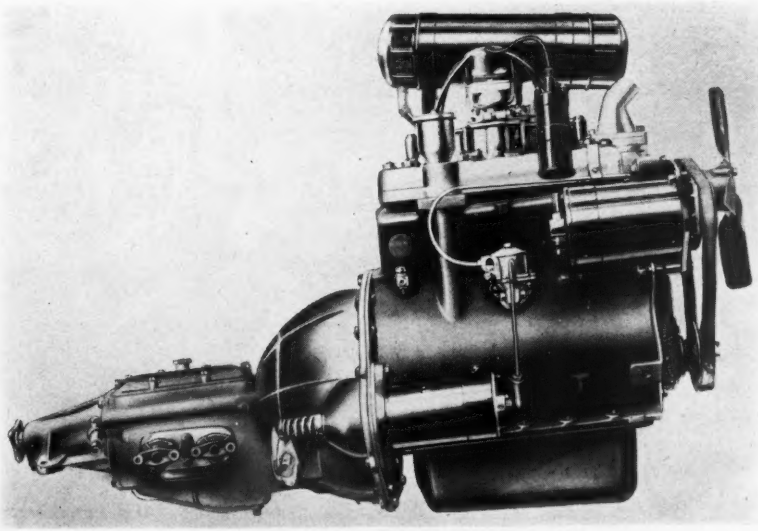
end of the crankshaft is furnished with two Woodruff keys for the timing sprocket and the fan-belt pulley; an oil thrower ring being clamped between them. Immediately in front of the oil thrower ring, housed in the pressed steel timing cover and bearing on a rearward tubular extension of the pulley, is a lip-type seal with a leather washer as protection from dust and grit. The wheel and sprocket assembly on the 1 in diameter shaft is secured by a nut incorporating dogs for the starting handle. Tabbed shims of differing thickness are available to lock this nut and to ensure the correct angular relationship between the starting crank and the crankshaft.

Stamped, I-section, En 8 R connecting rods with a centre length of $7\frac{1}{4}$ in are employed. The big-end bearing is split at right angles to the axis of the rod. Vandervell steel-backed, plain white-metal lined bearing shells are located in the big ends in the same manner as the journal bearings. Plain bolts, with a flat machined on their heads to prevent rotation, secure the bearing caps, and the slotted nuts are locked by split pins. Made of Clevite 10, the small-end bush is a press fit in the connecting rod, and has star-pattern oil grooving. End location of the gudgeon pin is by circlips in the piston bosses. The bosses are drilled to pass oil-splash to the bearing surfaces for the pin.

Supplied by Light Production Co. Ltd., and Wellworthy Piston Rings Ltd., the light alloy pistons have slotted skirts; the vertical slits in the pressure faces running from the base into a horizontal slit in the lowest ring groove, where it serves as an oil return. Three rings are fitted. The top one is chromium plated, the second has a tapered periphery, while the bottom one is a slotted scraper. Pistons are tin-plated for running-in and are matched to the bores by selective assembly.

End location of the four-bearing camshaft, of En 32B case-hardened steel, is by a semi-circular steel plate at the front end. It is $\frac{1}{4}$ in thick, and engages a groove machined in the camshaft. Just below this plate there is a pressed steel scoop, not shown in illustration, bolted to the crankcase wall, directing oil on to the crankshaft timing sprocket. The cams are of harmonic design. Helical grooves in the camshaft bearings provide for an intermittent oil feed through drilled ways from the gallery, except at the front bearing which supports the timing wheel. Here a circular groove coincides with the supply duct to ensure a continuous flow of oil to this bearing. The helical gear that drives the distributor spindle is integral with the shaft, and lubricated by splash.

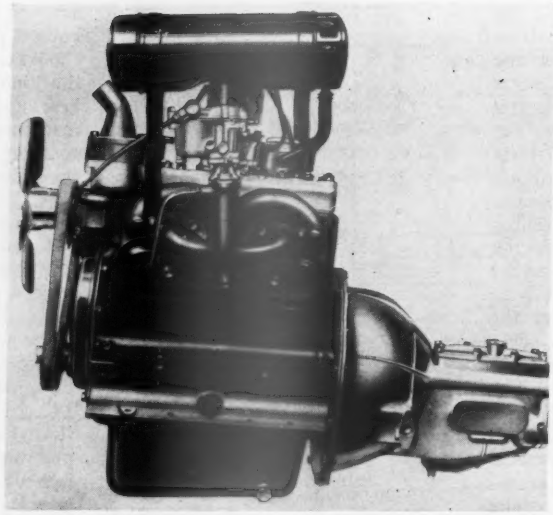
The camshaft driving wheel is attached by two bolts locked by tab washers. Four holes are drilled in the wheel for the two bolts and, by using the alternative pairs of holes, adjustments of half a gear tooth pitch can be made to the timing. By



Complete power unit, off side.

turning the wheel over, ¼ pitch adjustments can be obtained. The drive is by a single chain with a patented leaf-spring tensioning device. One end of this is secured by a bolt to the timing cover whilst the free end bears against the cover side wall. The spring is chromium-plated to increase wear resistance at its area of contact with the chain. Two leaves are used to provide a measure of damping.

The pressed steel timing cover is bolted on, together with the front engine plate which incorporates the engine mounting brackets. A stepped and waisted hexagonal stud gives additional support near the centre of the cover, and fibre washers prevent oil leakage. An external coating of Bittac reduces noise.



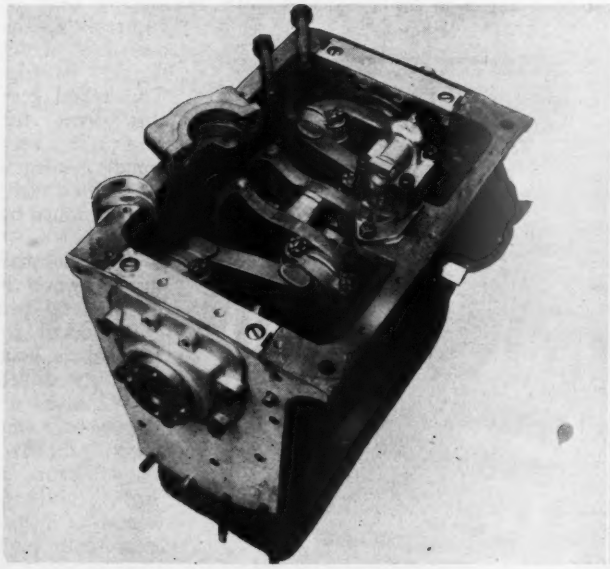
Near-side of power unit showing manifold arrangement.

Inlet valves open 10 deg before T.D.C. and close 50 deg after B.D.C. whilst exhaust valves open 50 deg before B.D.C. and close 10 deg after T.D.C. The valve lift is ¼ in and the tappet clearance, cold is 0.015 in for all valves. Single valve springs, secured by Holdsworth collars, are used and the loading is 22 lb when the valve is closed, and 37 lb when open. An included angle of 89 deg is adopted for the valve seats in conjunction with an included angle of 90 deg on the valves. Material for the exhaust valve is En 59 and for the inlet valve is En 52. The pressed-in, cast-iron valve guides have an internal diameter of 0.250 in, and are interchangeable. The valve port throat diameters are $\frac{7}{8}$ in exhaust, and $1\frac{1}{16}$ in inlet; the valve head diameters being 1 in and $1\frac{3}{16}$ in respectively.

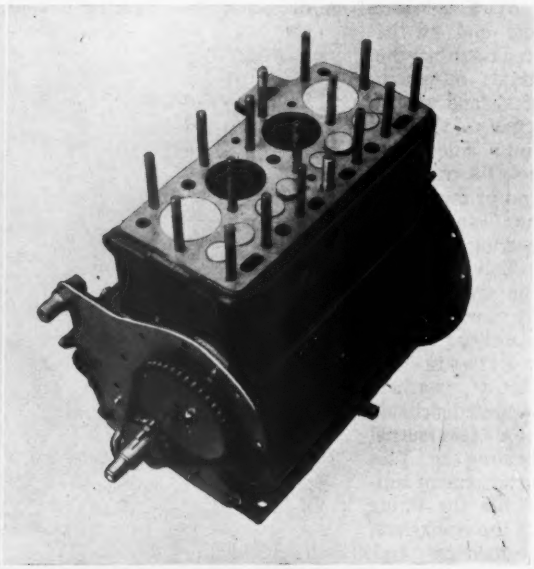
Cast iron tappet guide blocks are in pairs secured to the crankcase by two bolts, one between each pair of tappets. A cast iron bridge piece secured by two bolts to an abutment plate mounted across the gap between the two tappet guide blocks furnishes a support for the vertical drive shaft to take the thrust of the fuel pump drive spindle. At the same time, by bearing against the top face of the driving sleeve, it locates the spindle against upward movement caused by hydraulic pressure from the oil pump. The tappet cover is held in place by two internally threaded bolts on long studs. An oil-tight joint is made by a washer at the crankcase face. This washer also serves as a baffle to prevent oil being splashed up the breather pipe which connects the tappet chamber with the induction manifold. A disc with a small drilled hole fitted with a split pin is arranged at the manifold end of the pipe to prevent excessive air affecting the slow running of the engine. The breathing and crankcase ventilating system includes a pipe connecting the oil filler to the air filter.

The oil pump, distributor and fuel pump are driven by a sleeve on which is machined a skew gear, engaging with that on the camshaft. This sleeve, made of En 32B, also embodies the fuel pump actuating eccentric. Into the sleeve is pinned the spindle that drives the oil pump at its lower end and the distributor at the top in the customary manner. The flange of the phosphor bronze spindle bearing bears against the lower face of the sleeve to take the thrust of the skew gear drive. At the upper end of the spindle a slot engages a tongue on the sleeve-coupling driving the distributor. The usual eccentric tongue on the contact breaker driving-sleeve engages a slot at the top end of the sleeve-coupling to ensure correct assembly. Cast in N.F.17 aluminium alloy, the oil pump is spigoted and bolted to the lower face of the crankcase.

From a floating pick-up, fitted with



Crankcase inverted to show mounting of crankshaft and rear oil seal.



Cylinder block assembly with timing gear.

a gauze filter and deriving its buoyancy from a sealed chamber, oil is taken by the Hobourn Eaton pump and delivered to the main gallery. Pressure is maintained between 40 and 60 lb/sq in by a relief valve which bypasses excess oil to the sump in the usual manner. The main oil gallery, which is fitted with a pressure gauge connection, is drilled in the crankcase wall. A transverse drilling from the front end of the gallery connects with another drilling communicating with the front main journal and the front camshaft bearing. Elsewhere, individual drilled connections carry the oil to each of the remaining three camshaft bearings and to grooves around the other two main journals. Drillings to the camshaft are of a smaller diameter than the journals so that some proportional distribution is provided. Annular grooves in the housings and caps distribute oil around the journal bearings, and allow it to pass through a hole in each half of the bearing shell. Drilled passages in the crankshaft convey oil from the main journals to the crankpins.

The cast iron tappet guides are provided with an intermittent oil feed by means of helical grooves and flats on the second and third camshaft bearing surfaces. These connect with drillings communicating between the bearings and a straight groove running out at each end of the inner face of each of the guide blocks. This straight groove permits a continuous flow of oil, thus preventing the accumulation of sludge. Holes drilled in the tappet guide blocks pass oil from the groove to the chill-cast tappets. These are fitted with the usual case-hardened, hexagon-headed adjustment screw and locknut. S.A.E. 30 oil is recommended for Summer use, and S.A.E. 20 for Winter, 6 pints being the capacity of the sump.

Anodic treatment is applied to the cast aluminium cylinder head to protect it from corrosion which is particularly liable to occur in the stud holes. The sixteen holding-down studs are screwed into the top of the cylinder block, a clearance being machined round the tapped holes to avoid gas leaks caused by local distortion. Combustion chamber design is the same as on previous models, the squish effect, resulting from the piston crown coming very close to the similarly shaped head on the side remote from the valves, provides the requisite measure of turbulence. Adequate cooling passages are arranged over the combustion chamber and around the sparking plug boss. On the water outlet is the usual thermostat operating at 70-80 deg C.

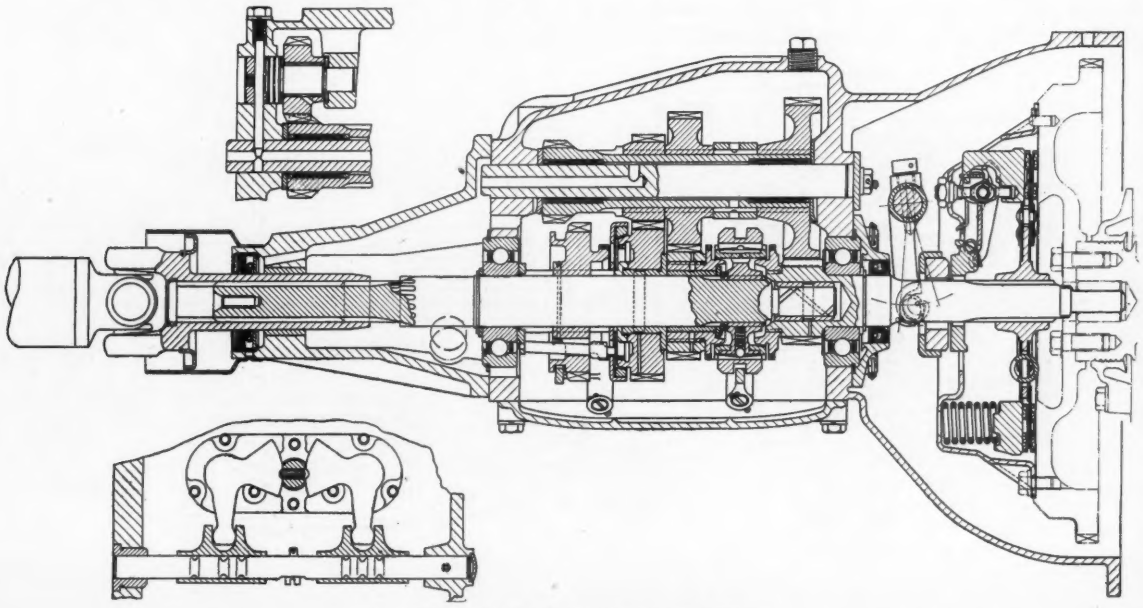
A water pump of the normal impeller type is employed. It is attached to the front of the cylinder block by two bolts and is driven by a V-belt from the crankshaft; the triangulated drive including the dynamo in the usual way. The pump driven spindle is mounted on two ball bearings carried in an extension of the pump body. On the forward end of the spindle is the driving pulley and boss, on the end of which the four-bladed, pressed steel fan is bolted. Unequal spacing of the fan blades reduces noise. Well spaced by a 1 in distance piece, the bearings are provided with a grease nipple on the pump body. Forward of the impeller assembly with its spring-loaded, rubber gland having a moulded-in graphite bearing face, there is a draining space and thrower to prevent any water leak from reaching the ball bearings.

For export an AC oil-bath air filter is fitted, whilst cars for the home markets are supplied with the AC gauze type filter and silencer. A cast iron inlet manifold having siamesed ports is cast integrally with the exhaust manifold and incorporates a hot spot. Two holding-down studs at the centre and one T-type stirrup at each end secure the manifolds to the block. A Solex downdraught carburettor is fitted, and an inlet manifold drain pipe is located at the lowest point. All joints are made with hard-bonded asbestos.

Lucas ignition components are standard equipment. The coil is mounted on the cylinder head, and ignition is set at top dead centre with the engine at rest, the customary automatic centrifugal advance and retard mechanism being fitted. Top dead centre is indicated when a $\frac{1}{16}$ in diameter hole drilled in the flange of the belt pulley on the crankshaft coincides with a pressed steel pointer on the timing cover. The sparking plugs are Champion N.A. 8.

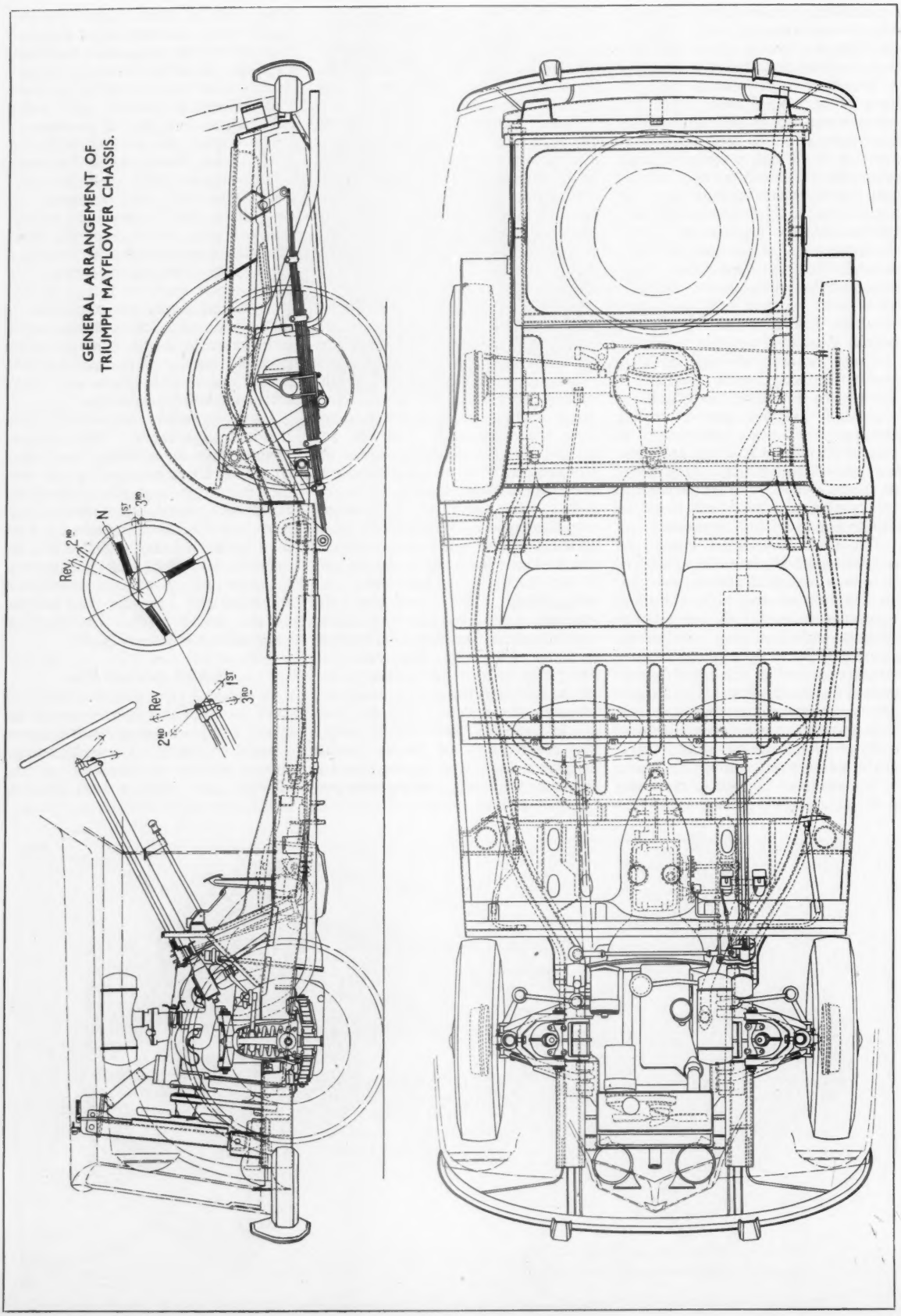
Clutch and gear box

A standard $7\frac{1}{4}$ in diameter Borg and Beck clutch, with a carbon withdrawal block, and a six-spring flexible centre plate, is employed. The pedal linkage incorporates a compensator of the pivoted and swinging arm type to



Arrangement of clutch and three-speed gearbox.

GENERAL ARRANGEMENT OF TRIUMPH MAYFLOWER CHASSIS.

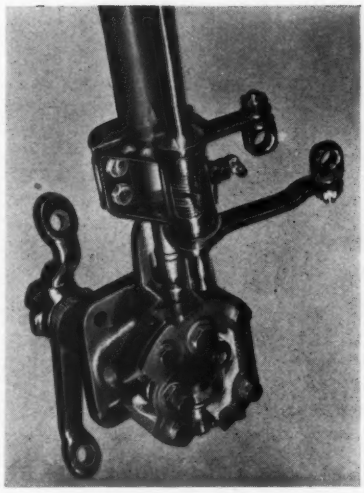


accommodate movement of the engine in a fore-and-aft direction.

Three forward speeds and reverse are provided by the gear box of conventional layout, the ratios being 1 : 1, 1.67 : 1, 3.54 : 1, and 4.11 : 1, giving the overall ratios quoted in the specification panel. Apart from some minor modifications to the gear box casing, the mainshaft, and the constant pinion shaft to suit the smaller car, the gear box is identical with that of the latest Standard Vanguard. The 1st and 2nd speed gears have a helix angle of 25 deg whilst the constant mesh gears are 30 deg. The normal diametral pitches of the gears are: 1st speed 9.337, 2nd speed 11.034, and the constant mesh 11.547, whilst the diametral pitch of the straight spur reverse train is 10. All gears are made of En 24, except the first layshaft gear with its integral sleeve which is of En 325.

In view of the fact that propeller shaft movement is not so restricted on the Mayflower as on the Vanguard, the rearward extension of the gear box is comparatively short. It houses the sliding joint, which is supported in an Oilite bush, and the speedometer gears. A recess to the rear of the bush carries a lip-type oil seal incorporating a protective leather ring. A drilled hole provides a drain between the seal and the bush. The seal is further protected by a thrower ring shrouded in a pressed steel sleeve at the rear end of the gear box extension.

The control rod runs from the top of the steering column to the actuating gear at the base in the usual manner. Instead of the dog and slot arrangement used on the early Vanguard gear controls, it carries at its lower end a peg that engages in one or other of the slots formed in the two sleeves welded to the operating levers. These sleeves are machined to form screw type bearings in the casing of the selector mechanism. The whole assembly is grease-lubricated through a nipple on the casing. Engagement of the upper sleeve gives reverse and first gear, and selection of second and third is made in the lower one. Motion is then transmitted by rods downwards from the operating levers to levers mounted on two cross-shafts, adjacent to the ends supported in spherical self-aligning bearings on a frame bracket. At their other ends the cross shafts are connected by flexible couplings to the gear box. A spherical-ended pin, in conjunction with two notched sector plates and levers in the gear box, ensures that only one gear at a time can be engaged.



Gear selector mechanism.

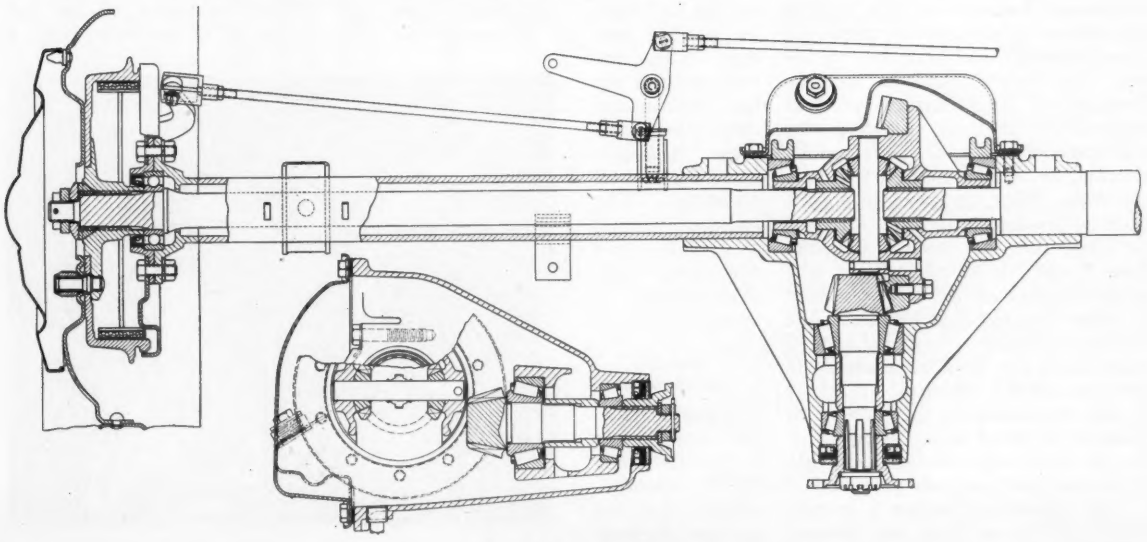
Full details of the gear box were given in the description of the Standard Vanguard in the September, 1948 issue of *The Automobile Engineer*.

Rear axle

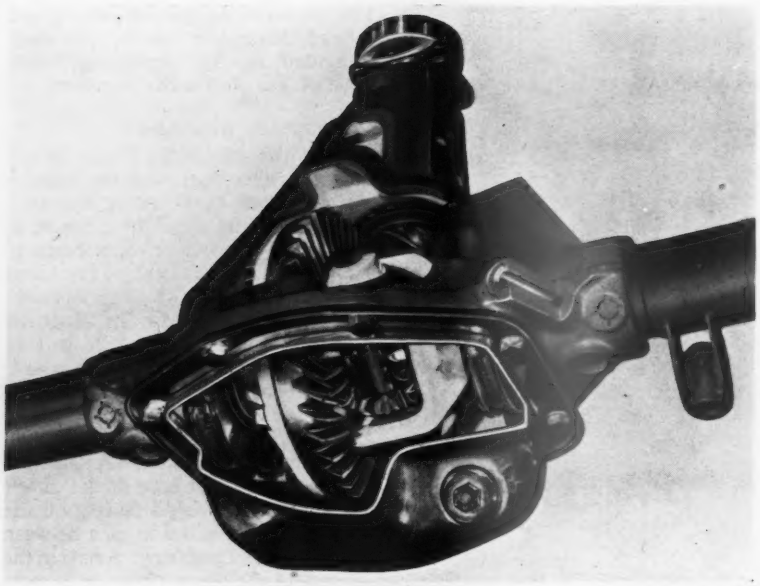
From the gear box a Hardy Spicer open propeller shaft takes the drive to a hypoid gear, semi-floating rear axle. Support for the main drive pinion is provided by two taper roller bearings housed in the malleable cast iron differential casing. The pinion, formed integrally with its shaft, is overhung and its centre line is 1 in below the axis of the crown wheel. At the forward end of the casing there is an oil seal bearing on the boss of the universal joint flange, which is splined on to the pinion shaft. A large oil passage cast in the nose of the differential casing allows oil flowing off the rotating crown wheel to pass between the taper roller bearings. A hole in the bottom of the bearing housing permits a continuous flow of oil through the space between the bearings.

The crown wheel, which is of somewhat greater width than is necessary to provide the 1 in length of teeth, is secured on the one-piece, malleable cast iron differential cage by ten $\frac{15}{16}$ in bolts locked with tab washers. The differential is of the two-pinion type, the pinions being mounted on a $\frac{5}{8}$ in diameter spindle and bearing at their outer faces on dished phosphor bronze thrust washers. The offset towards the crown wheel of the differential pinion shaft from the centre line of the main drive pinion is 0.38 in.

Internally splined differential gears carry the inner ends of the half shafts which are reduced locally from $1\frac{5}{32}$ in to $\frac{31}{32}$ in diameter. The cage is mounted on taper roller bearings



Semi-floating rear axle.



Hypoid final drive, with two-pinion differential.

secured to the casing by caps, bolted-on and locked by tab washers. Holes in the differential wheels allow oil to pass to their thrust washers, and holes in the cage provide for lubrication of the bearing surfaces. The crown wheel and differential pinions are of En 35 A leaded steel, and the drive pinion is of En 325 steel. A pressed steel cover incorporating a filler plug is bolted to the rear face of the differential casing in the base of which a drain plug is provided. S.A.E. 90 hypoid gear oil is used, the correct level being indicated when the oil is up to the lowest thread in the filler hole.

The first adjustment on assembly is the fore-and-aft location of the main drive pinion. The position is determined with the aid of a jig, and adjustment is effected by means of shims between the outer race of the rear taper bearing and its housing. Gauging of the bearing pre-load applied by the nut securing the universal joint flange is effected by checking the torque necessary to turn the shaft. Next, the casing is sprung open a pre-determined amount by a jig fitting into the two $\frac{5}{8}$ in diameter holes in the axle tube housings. The complete pinion cage assembly with its roller bearings is inserted into the differential casing. Behind the inner races shims are fitted on assembly to give the correct pre-load. A gauge is used to determine the number of shims to be fitted before assembly so that no time is lost as would be the case if trial and error methods were used.

Two tubular extensions 2 in diameter and $\frac{3}{16}$ in thick, are pressed into the differential carrier and pegged. They have stamped flanges electrically butt welded to their outer ends. Secured to these flanges, by four bolts with spring washers, are the brake back plates and the outer half of the bearing housings that also carry the oil seals. They are fitted with grease nipples, for the ball bearings. The usual brackets for the springs and brake controls are provided on the axle tubes.

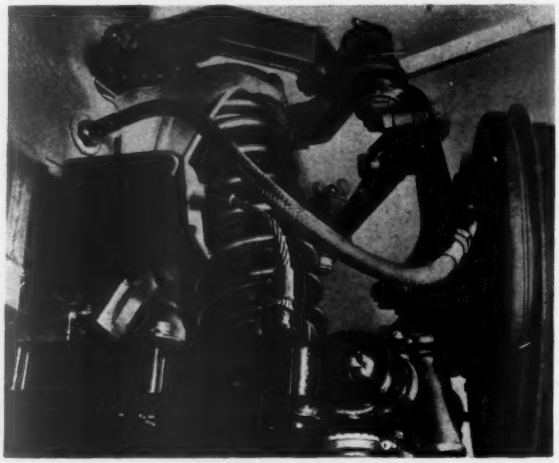
The internally splined wheel hub has a conical ring seating in a suitable recess in its outer face and is attached to the half shaft by means of a split-pinned nut and thick washer. Lateral location of the hub is provided by the inner race of the ball-bearing, which in turn is in contact with a shoulder on the half shaft. Gripped between its housing and the flange of the axle tube, the outer race of the ball-bearing locates the complete half-shaft and the wheel assembly. The standard method of wheel and brake drum attachment is used.

Front suspension

Fundamentally, the arrangement of the telescopic shock absorber within the coil spring of the front suspension is very sound. The line of action of the shock absorber loads coincides with that of the spring, thus simplifying the structure needed to take these loads. Furthermore, a considerable saving of space is obtained, and the top wishbone arms and supports can be made lighter and more inexpensively than is the case when they have to actuate the shock absorber. Wide spacing, $10\frac{1}{8}$ in, minimizes the loading of the top and bottom wishbone inner bearings, which are positioned 12.66 in and 9.31 in respectively from the centre line of the vehicle. The centre length of the top wishbone is $6\frac{1}{2}$ in whilst that of the lower one is $10\frac{1}{4}$ in. Camber and castor angles are 2 deg and zero respectively and a toe-in of $\frac{1}{8}$ in is used, whilst the inclination of the swivel pin is 9 deg.

A fabricated 16 s.w.g. pressed steel bracket welded to the frame houses the top end of both the coil spring and the shock absorber which is attached by means of the usual rubber and steel sandwich fitting. The lower end of the shock absorber is attached by the usual ring with conical rubber bushes. This permits a fair degree of angular movement and helps to insulate the structure from noise and vibration. Further refinement is provided by mounting both ends of the springs on rubber. As can be seen in the illustration, a rather complicated arrangement of bolted-on, pressed steel brackets supports the lower end of the spring and shock absorber and permits the withdrawal of the shock absorber without dismantling the whole suspension unit.

Supported on a one piece steel stamped fulcrum machined at its ends to take the bearings, the two halves of the top wishbone are held together by a single slotted nut on the threaded and splined extension of the ball joint at the top of the swivel pin. A



Front suspension with bump stop and rebound check cable.

distance piece is inserted in the wishbone to support it at the outer end where the top of the inverted U-section is cut away so that the resulting fork ends on the arms of each half of the wishbone may overlap to form the junction. Rubber bushes in tubes welded to the inner ends of the wishbone are used for the inner fulcrum bearings. The top wishbone is pressed out of two straight pieces of 12 s.w.g. which have been mitred in the blank form in two places so that they can be bent to shape, and welded after a straight pressing operation. A reinforcing plate is welded to the bottom of each arm between the inner fulcrum and the first bend, thus boxing in that part of the section. The $1\frac{1}{32}$ in diameter ball portion of the outer joint is bolted to the top of the swivel pin. It is self-adjusting for wear, lubrication being provided by a grease nipple at the top, and the ingress of dust is prevented at the bottom by the usual rubber gaiter.

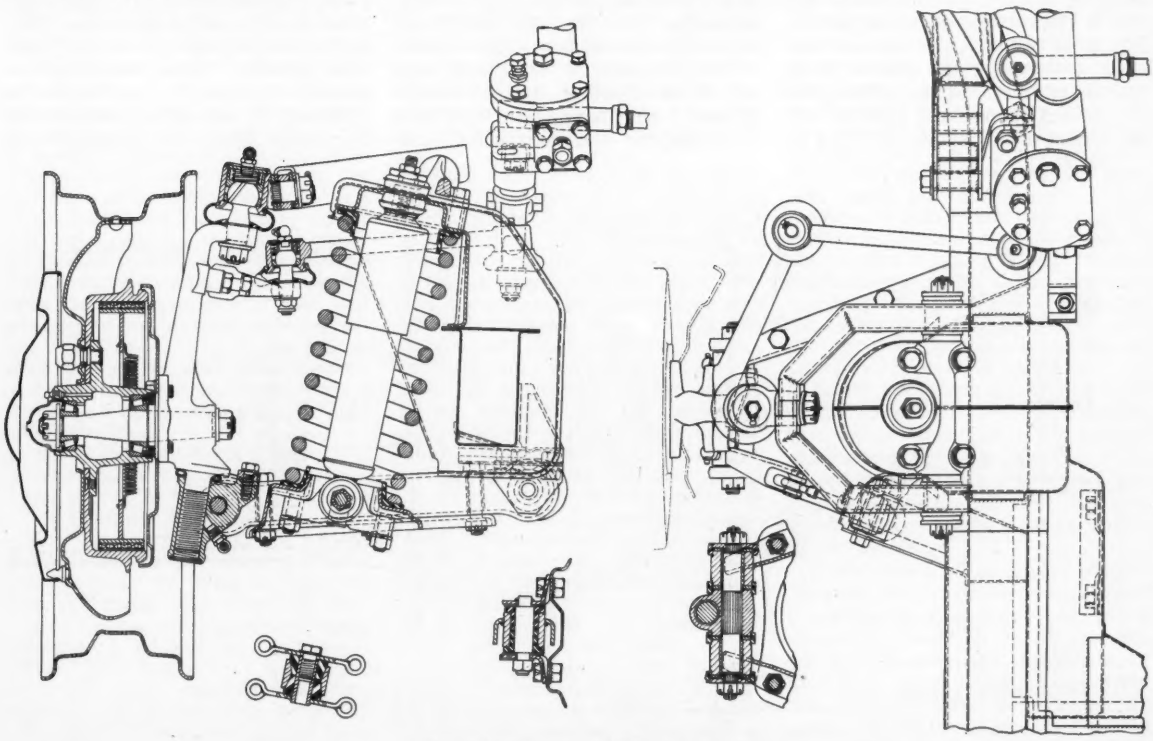
The 10 s.w.g. lower wishbone arms each consist of a pressed C section with welded-on tubular end fittings housing the bearings. Rubber bushes for the inner bearings are supported in removable 14 s.w.g. bearing lugs bolted to the frame. At the outer end, bearings fitted with star-grooved, Clevite 10 bushes are mounted on a spindle splined into the forging which forms the lower screw-type bearing for the swivel pin. Rubber rings are used to cover the joints of the wishbone bearing to keep out dust and mud. Clevite washers on each side of the bearings take the end thrust, and sealing is completed by washers splined on to each end of the spindle. The lower swivel pin bearing is sealed at its base by a peened-on cap and at its top end by a C-section rubber seal. Three grease nipples are provided for the lubrication of these bearings.

Full bump loads are taken by a 12 s.w.g. bracket attached to the forward face of the spring mounting bracket and to the chassis side member. Striking this bracket is a rubber stop mounted in the centre of the forward lower wishbone arm, an arrangement which must, at times, induce some comparatively severe stresses in this member. A fork joint, pivoting on a bolt on the outer end of the same wishbone arm, and a trunnion on the bump stop bracket provide for the attachment of a cable which supports the wheel on rebound. Total front wheel movement from bump to rebound is $6\frac{1}{4}$ in, the effective spring rate being 81.25 lb/in giving a car front end frequency of 74 cycles per minute.

The forged swivel pin, in addition to the end bearings already mentioned, has two lugs about $4\frac{3}{4}$ in above the wheel centre line. To these is bolted the steering lever, the outer end of which carried the usual ball joint. A noteworthy point is that the spring-loaded, self-adjusting part of this bearing is a steel pressing, as compared with the similar part at the top of the swivel pin which is machined from phosphor bronze. Four bolts attach the brake back plate, and a split-pinned, slotted nut and washer secures the stub axle in a tapered hole in the swivel pin. Another split-pinned, slotted nut and washer bears against the inner race of the outer taper roller bearing supporting the hub ; the inner race of the inner bearing abutting a shoulder on the stub axle. Lubrication is by grease nipple, a felt seal at the inner end, and a pressed steel cap over the hub retaining nut at the outer end, keep the grease in. The usual method of wheel attachment is used.

Rear suspension

Eight leaves $1\frac{1}{2}$ in wide and four tension plates form each of the Toledo Woodhead semi-elliptic springs which are spaced 36 in apart. The bottom leaf, 0.188 in thick, is plain but the remaining ones are grooved to provide greater flexibility without any increase in weight. Three of the grooved leaves are 0.208 in thick, and the top four have a thickness of 0.231 in. Handling characteristics of the vehicle are improved by the positioning of the front spring attachment $5\frac{1}{4}$ in below the rear shackle, and of the axle $20\frac{1}{4}$ in from the front attachment, or $25\frac{1}{2}$ in from the rear shackle pin. Static spring deflection is 6.6 in, and the



General arrangement of front suspension.

spring rate of 104 lb/in gives a natural frequency of 73 cycles per minute.

The centre-line of the pressed-steel spring anchorage bracket is some 3 in offset from the centre line of the frame side member to which the bracket itself is welded, and further support is provided by a cross member extending from the side member inwards to the tunnel. A pressed-steel bracket on an extension of the frame side member carries the shackle at the rear. Rubber bushes are used throughout.

Each end of the telescopic shock absorber incorporates a ring fitting used in conjunction with a rubber bush. At the top end it is attached to an overhung trunnion mounted on a bracket on the side member where it joins the rearward frame extension. Shock absorber anchorage to the axle is by means of a trunnion bracket welded to the plate for the U-bolts securing the spring. No rebound stop is fitted but a rubber pad on the top of the axle in conjunction with the frame side member, forms the bump stop.

Steering

Unobstructed vision of the dash instruments is ensured by the arrangement of the T-spoked sprung wheel. The cam-and-follower steering box, lubricated with S.A.E. 90 E.P. hypoid gear oil, is bolted to a bracket arc-welded to a gusset between the frame side member and the toe-board. Actuation of the outer tie rods and track rod is by a forged lever pivoted near its centre on the operating spindle. The other end of the track rod, which passes to the rear of the cylinder block, is attached to an exactly similar idler lever which operates the other tie rod. Movement of the wheels is 25 deg on front lock, and 31 deg on back lock, with an overall steering wheel movement of $2\frac{1}{4}$ turns, giving a turning circle of 34 ft.

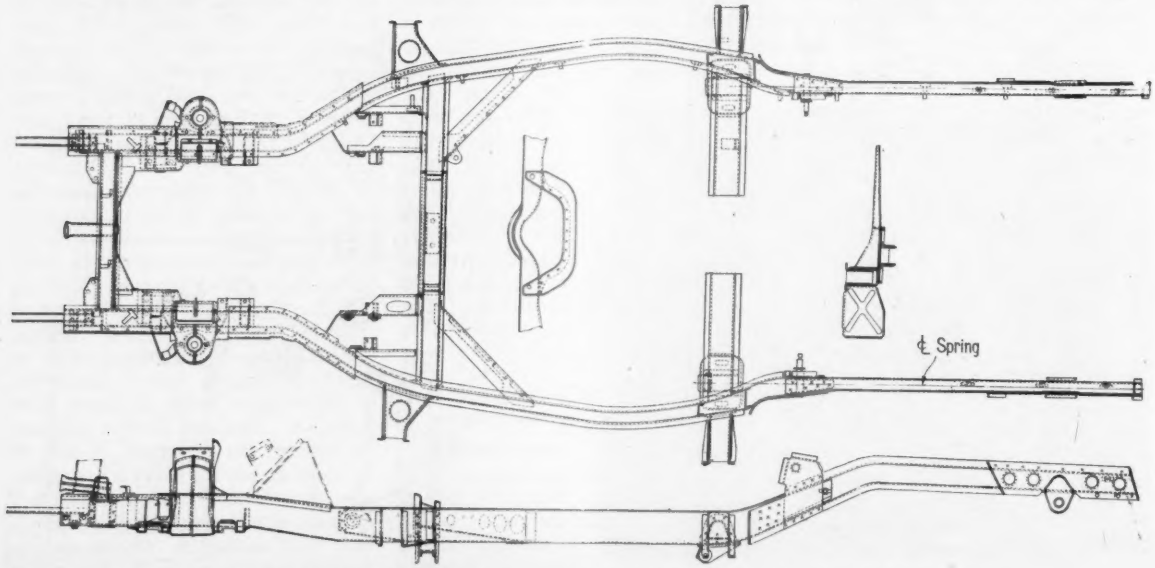
Rear brake compensator.

Frame

Perhaps the most remarkable feature of this frame is the complete absence of the usual cruciform bracing and rear cross member, their functions being performed by the body and the floor pressing. In fact, there is only a 3 in $\times$ 2 in box-section front member and one other cross member, of inverted top-hat section of varied depth and braced by two diagonal channel-section stays back to the frame. Under this cross member is bolted a cradle to carry the rear engine mounting. All members are made of 18 s.w.g. pressed steel.

Brackets extending forward from the cross member and supported on the side members where they sweep inwardly, carry the pedal shafts and stops and also the gear control shafts.

Over the centre portion, the 18 s.w.g. side members are of inverted top-hat section 4 in deep $\times$ $1\frac{1}{4}$ in wide plus $\frac{5}{8}$ in flanges; the floor being spot-welded on to complete the box. Where the side members sweep in to clear the front wheels on lock, the top of the box is formed by a separate 18 s.w.g. welded-on plate. An additional, reinforcing, box section carrying the front engine mounting bracket is spot-welded on the inside of the side members forward of the spring bracket and well gusseted to the 18 s.w.g. front cross member. A bell-mouthed tube is welded on to the front member to guide the sturdy starter handle; a simply but highly desirable feature. The T-shaped fittings shown in the illustration are wiring lugs. Rearward extensions of the frame, supporting the rear spring shackle brackets and the boot, are of 18 s.w.g. top-hat section $1\frac{3}{8}$ in $\times$ $2\frac{1}{2}$ in, plus $\frac{5}{8}$ in flanges. They are completely boxed by the inner vertical face of the 20 s.w.g. wheel arches and by additional plates at the extreme rear. Four plates reinforcing the jacking brackets on the body sills are arc-welded to the sills at their outer ends, and to the frame side members at their inner ends. One pair is adjacent to the rear engine cross member while the other pair is in line with the front anchorage brackets of the rear springs. As mentioned before, the front spring brackets are welded to pan-shaped pressings, which are in turn arc-welded to 18 s.w.g. semi-cross members. These are of a shallow inverted top-hat section where they pick up with the edge of the propeller shaft tunnel but deepen to a more arched section to give a good spot-welded attachment to the frame side member. This rather unusual shape is necessary to conform to the profile of the foot wells pressed in the rear of the floor. The propeller shaft



Basic frame to which the body is welded. Details: Gearbox support and rear spring anchorage.

tunnel only extends about 18 in forward from the heel-board and in general, apart from a top-hat section cross member supporting the front seats and the rear foot well, the swaged floor is flat.

Other features

Lockheed hydraulic brakes of a standard type are used with two-leading shoes on the front wheels and leading and trailing shoes at the rear. The brake drum diameter is 8 in, with shoes of 1½ in width, giving a friction lining area of 90 sq in. Hand brake operation is by a pistol grip handle. A Bowden cable carries the motion to a cross lever mounted on one of the channel-section frame bracing members. This lever, 9¾ in long, multiplies the applied force in the ratio of 3 : 1, the increased load then being transmitted to a compensator mounted on the rear axle. Compensation, that is to say equilization of the brake rod loads, is accomplished by a rod bent to a right angle; each end being machined to form a screwed journal. One end is mounted in a screw bearing on the axle and on the other end the brake operating lever, furnished with another screw bearing, is accommodated. Grease nipples are provided for the lubrication of these bearings and felt seals give protection from dust.

Charging of the Lucas 12-volt 38 amp-hr battery with positive earth return, is regulated by a compensated voltage control with a cut-out. Separate head and side lamps are fitted, as well as twin tail and brake lamps with separate rear number plate illumination. Other electrical services include the interior lamp, ignition warning light, panel light, direction indicators, starter motor, windscreen wipers, and a pair of Windtone horns.

The fuel tank, mounted in the body, holds ten gallons, and the coolant capacity of the finned-tube radiator is twelve pints. Drain taps are provided on the right-hand side at the rear of the cylinder block and in the radiator bottom tank. A water temperature gauge is fitted to the system. The spare wheel is housed in a cradle immediately below the fuel tank. Pivoted at its forward end, the cradle has a latch at the rear locked by a quick-release bolt passing through the floor and secured by a nut inside the boot.

An almost straight run is provided for the exhaust system, which incorporates a silencer positioned longitudinally under the floor. Three rubber sandwich mountings are used; one at the rear engine support cross member, one near the top of the arch over the rear axle, and the other at the extreme rear end of the pipe.

Heating and ventilating was effected by a Smith's 2 kilowatt rectangular unit, which takes in air from the scuttle ventilator, or can be used as a re-circulator. Later models, however, are fitted with a 3 kilowatt unit. A Smith's push-button radio is also supplied as a standard extra. In general, comfort and refinement have been well considered in the design of this car. Rubber insulation has been used to advantage to eliminate road noise, and at both front and rear of the engine rubber-to-metal bonded sandwich mountings are used. Furthermore, insulation of the body structure from engine vibrations is even more complete owing to the fact that no longitudinal tie-rod is used. This is made possible by the incorporation of the compensating link in the clutch control system. In spite of the unusually close relationship between the natural frequencies of the front and rear suspension, pitching vibrations are adequately controlled by the telescopic shock absorbers.

AIR AND OIL FILTERS

VOKES LTD., Guildford, Surrey, have issued a report dealing with tests carried out to determine the effect of Vokes air and oil filters on a conventional diesel engine. This report follows the lines of that covering similar tests carried out with a spark ignition engine, and dealt with in the *Automobile Engineer* for January, 1948. The main difference, apart from the fact that a diesel engine was used in this case, was that the dust feed to the air intake was regulated to simulate conditions under which the majority of diesel engines have to operate, whereas in the earlier test, a higher rate of dust feed was used to provide most severe operating conditions. The investigations were designed to show what happens to a conventional diesel engine when it is run in a dusty atmosphere for 120 hour test periods :—

(i) without any filters fitted.
(ii) with a Vokes air filter fitted.
(iii) with a Vokes air filter and a Vokes oil filter fitted.

The tests were carried out by the Moore and Barrett laboratories, Whyteleafe, Surrey. By fitting a Vokes air filter the engine was protected from most of the harmful action of the dust, and with the addition of a Vokes full flow oil filter, engine wear was further reduced by quite a considerable amount and operating efficiency was improved.

The air filter used was supplied from the Vokes standard Micro-Vee range and consisted of a corrugated felt filtering element supported by wire gauze and housed in a cylindrical louvred casing. A Vokes standard full flow type oil filter was used, which consists of a corrugated felt filtering element in cylindrical form supported by wire gauze and fitted with metal end-caps.

Piston wear (based on maximum decrease in diameter) is reduced by 50 per cent when the air filter is fitted to the engine. No further saving in piston wear resulted from the addition of the oil filter. Cylinder bore wear (based on maximum increase in diameter) is decreased 70 per cent. by the air filter and a further 12 per cent when the air and oil filters are both in circuit. Total loss in weight of all five rings is 1·4740 grammes on the unfiltered engine, 0·5489 gramme when the air filter is fitted and 0·3382 gramme when both the air filter and oil filter are in use. The reduction in total ring wear is therefore 63 and 77 per cent respectively. Measurement of the loss in weight of piston rings should be considered the most reliable because such figures can be more accurately determined than linear measurement of piston and cylinder diameters.

The fitting of the air filter did not materially affect valve stem wear but the addition of the oil filter reduced the wear 100 per cent. This result is based on measurements accurate to a ten-thousandth of an inch. The total loss in weight of both bearing shells was 0·7703 gramme without filters, 0·1905 gramme with the air filter and 0·1013 gramme with the air filter and the oil filter, giving savings in bearing wear of 75 and 87 per cent respectively. In the crankshaft journals the oil filter brought about a saving in wear greater than was effected by the air filter. The maximum decrease in journal diameter is 0·0008 inch without filters, 0·0005 inch with the air filter and no measureable wear could be recorded when the air and oil filters were used together. Based on measurements to the nearest ten-thousandth of an inch, the respective reductions in wear are therefore 37·5 and 100 per cent.

Photographs showing magnified views of the respective oil samples confirm that the air filter removes substantial quantities of the large particles, whilst the addition of the oil filter not only eliminates further particles but leaves a reduced concentration of impurities of smaller size.

If the speed of an engine remains constant, then oil consumption is closely associated with cylinder and piston assembly wear. These experiments confirm this statement. Use of the air filter reduced oil consumption 25 per cent. The air filter and oil filter together reduced consumption by 44 per cent.

WEIGHT CONTROL

Some Considerations on the Design of Lighter Cars

IT was we believe, Henry Ford I who first priced a motor car by weight. He it was also, it is said, who then priced his car per lb at the prevailing price of beef. Much development has however taken place since the design of the Ford model T, and the relatively comparable car is now considerably heavier and more complicated. Today, excluding luxury vehicles, the majority of cars conform roughly to an average minimum weight, which although higher than the model T power for power, seem nevertheless to be broadly set at a prevailing economic level. In other words, it seems unlikely that appreciable weight saving would be accompanied by a proportionate decrease in price.

In fact, if the weight reduction were to be obtained by the extensive use of light alloys, and the introduction of additional machining operations, the price might well be increased. It should therefore be the aim to economise in the use of materials by increased efficiency of design, and by a very careful investigation of the weight aspect. Such an investigation cannot be carried out by the designer who has other important problems requiring his time and attention, but it must have the undivided attention of a specialist if results, better than those obtained from a mere cursory examination of the problem, are to be obtained.

The tabulated figures are the overall dimensions and kerb weights, and figures are given for the specific weight, or weight per unit volume. It must be assumed that each car has been proved, on test and in use, to be strong enough and stiff enough for the purpose for which it was designed. The specific weight is therefore an indication of the amount, or rather the density of material used, to attain these conditions. On this account, it is also an indication of the structural efficiency or strength weight ratio. Cars in the luxury class and those of higher price will embody more amenities, and hence will be weightier than the less expensive ones. For that reason some way of presenting the tabulated figures must be devised in order to distinguish between cars in different price ranges. The graphical method of presentation appears to be the most suitable, as it shows clearly how the specific weight increases with the size of car. At the same time, it forms a basis for comparison of the policies of different manufacturers.

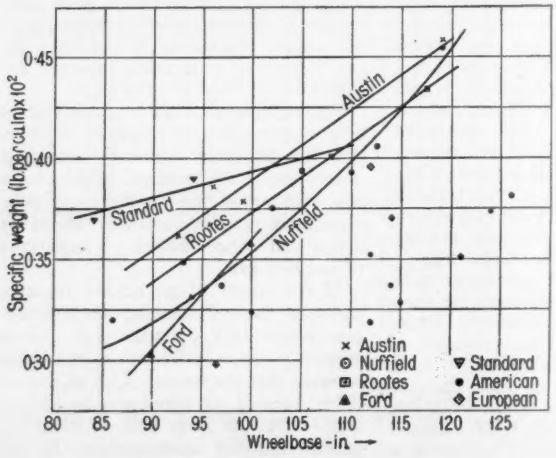
Specific weight is plotted against wheelbase on the assumption that the value of quantity produced cars is a function of the wheelbase. The word value has been used here as opposed to price, since price is a very arbitrary parameter which depends upon, among other factors, the marketability of the product and also of other products of the same firm. The arbitrary nature of the price of cars is the reason for the scatter of the points about the straight line curve of basic price against wheelbase. Points on this graph are taken from the popular priced range of Austin, Morris, and Rootes cars.

Weight-cost curves

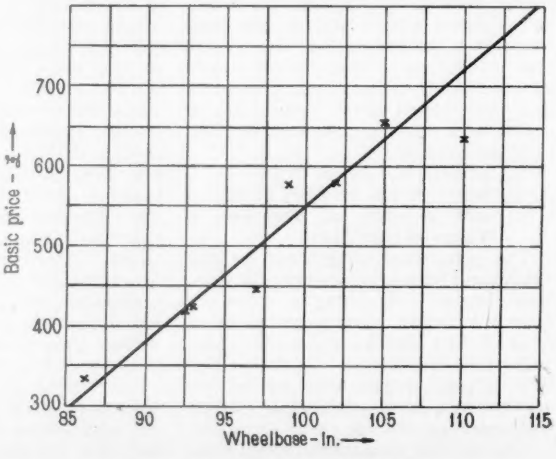
On examining the first mentioned curves, it is very interesting to note how consistent are each of the separate policies of the Austin, Rootes and Nuffield Companies. With regard to the Standard and Ford Companies, it is obviously wrong to compare the Mayflower and Triumph with the Vanguard, and the older styles of the Prefect and Anglia with the newer Consul; this has influenced the slope of the curves for these two makes.

Pre-war styles are conducive to low specific weights, as can be seen from the points plotted for European cars, most of which retain the old conventional front end to a greater extent than our own cars. What is most interesting is that all the American cars have very much lower specific weights than the British. It is well known that American cars are low in cost by comparison with our own, and it is generally assumed that this is largely due to greater quantity production. From these curves however, it can be seen that this is not the whole of the story. As has been mentioned before, material and its fabrication costs something like 3s. 10d. per lb, and it follows that a reduction of specific weight to a value approaching that obtained in American cars would help to close the price gap.

As regards the all important matter of saving weight, this can only be done by first introducing into the design office an awareness of the vital importance of weight saving. Every designer instinctively saves the odd



Specific weight related to wheelbase



Basic price related to wheelbase

DIMENSIONS AND WEIGHTS OF BRITISH CARS

Vehicle	Curb weight lb	Overall length, L in	Overall width, W in	Overall height, H in	Wheel-base in	Weight / L.W.H. × 100 lb/cu in
Austin A40 Devon	2128	153	61	63	92	0·362
„ A70 Hampshire	2800	163	67	65	96	0·385
„ A70 Hereford	2870	167	70	65	99	0·378
„ Sheerline	4300	192	73	67	119	0·457
Morris Minor	1736	148	61	60	86	0·320
„ Oxford	2296	167	65	63	97	0·335
Wolseley Four-Fifty	2660	170	66	63	102	0·375
„ Six-Eighty	2884	177	66	63	110	0·392
Riley 1½ litre	2695	179	63	59	112	0·405
„ 2½ litre	3170	186	63	59	119	0·454
Hillman Minx	1995	157	61	60	93	0·348
Humber Hawk	2870	174	70	64	105	0·368
„ Super Snipe	3843	187	72	66	117	0·433
Standard Vanguard	2845	166	69	64	94	0·388
Triumph Mayflower	2144	156	62	60	84	0·369
„ Renown	2984	179	64	65	108	0·400
Ford Anglia	1636	154	56	63	90	0·302
„ Prefect	1808	155	56	63	94	0·331
„ Consul	2415	173	64	61	100	0·357
Vauxhall Wyvern	2093	165	62	65	98	0·314
„ Velox	2278	165	62	65	98	0·342
Rover 75	3083	178	65	63	111	0·423
Jowett Javelin	2290	168	60	60	102	0·378
Singer SM1500	2520	174	63	64	107	0·360

penny here and there by simplifying production processes. In the same way he should instinctively save the odd ounce here and there to reduce the overall weight. These small quantities although insignificant by themselves, add up to a considerable total over the whole vehicle.

Having introduced into the drawing office the right attitude towards weight saving, it is then necessary to study the subject from an engineering point of view. There are many ways in which weight may be saved on an engine such as the substitution of pressings for castings where appropriate, and the use of aluminium instead of steel. These however are detail considerations that can only be decided on the drawing board for each individual design. A multiude of individual factors must influence final decisions. A word of warning is necessary in connection with the use of aluminium. It is not as stiff as steel, and in view of this fact, components made of this material must be carefully designed to avoid distortion.

Probably the greatest reduction in weight can be made on the actual structure of the car. There are three basic forms now generally adopted. The first is the conventional separate chassis and body type of construction. Then there is the unitary structure which is in effect a chassis frame welded to, and forming part of the body structure. The most advanced form is chassisless construction, which has virtually no frame at all. It is desirable to make careful distinction between the three types of layout.

The lightest structure

Fundamentally, the lightest structure is provided by chassisless construction. A frame as such, must be limited to a comparatively small depth. Hence it must be heavily constructed to be both stiff enough and strong enough. In the chassisless scheme advantage can be taken of the great depth of the vertical panels to take bending loads. Strength in bending varies as the square of the depth, and stiffness as the cube of the depth. It follows therefore that a body panel, which can be four times the depth of a chassis frame, is potentially sixteen times as strong and sixty-four times as stiff. This means that in order to carry the same load, the gauge of the body panel can be very much reduced as compared with the chassis frame. Much the same principles apply where torsional strength and stiffness are concerned, for strength is proportional to the cross sectional area enclosed by the skin, and torsional stiffness is proportional to the square of the area.

In actual fact the gauge cannot be reduced to the extent indicated by these figures, owing to potential buckling and instability. The 20 S.W.G. sheet steel commonly used in body construction is, in most parts of the car more than sufficient to provide the necessary structural strength and stiffness.

In general, quite an amount of weight can be saved by punching lightening holes in brackets, panels and frame members in places where the metal is not required. The cost of this is not very great, and the cut out material is sold for scrap. Lightening holes have been used more and more of recent years, but there is still plenty of scope for further extension of this practice.

Mechanical components can be pared down in much the same way as can engine parts, that is to say, by removing all metal that is not genuinely needed, as opposed to paring down the part all over and making it too weak. It is not suggested that extra machining operations should be introduced *ad lib*, but if the reduction of weight is worth while, the possibility of extra machining at the same time, and in the same jig as more essential operations should be considered. Weight reduction can of course be made in castings at the design stage at no extra cost.

Attention to detail is necessary, and for that reason the employment of a man whose sole job it is to keep a check on weights is worth serious consideration. The economics of such a proposition must of course be studied. Assuming £500 per annum

DIMENSIONS AND WEIGHTS OF FOREIGN CARS

Vehicle	Curb weight lb	Overall length, L in	Overall width, W in	Overall height, H in	Wheel-base in	Weight / L.W.H. × 100 lb/cu in
Nash Rambler	2450	176	73	59	100	0·323
„ Statesman	3040	201	77	62	112	0·318
Chevrolet Styleline	3170	198	74	66	115	0·328
Cadillac Series 62	4070	215	79	63	126	0·381
Ford Tudor	3080	197	73	63	114	0·337
Studebaker Champion	2770	190	69	60	112	0·352
Hudson Commodore	3590	208	77	60	124	0·374
Renault 4 C.V.	1332	142	56½	57	82	0·294
Citroen 6 cylinder	2912	189	70	61	121	0·351
„ Light Fifteen	2455	170	65	60	114	0·370
Lancia Aurelia	2490	174	61	59	112	0·396
S.A.A.B. 92	1680	155	65	56	97	0·298
Fiat 1400	2552	167	65	59	104	0·398

for the salary of the weight estimator, only one man is employed in most manufacturers design offices. The number of drawings issued per day is a good basis for estimating the number of men required for this work. In 30 years of service, £15,000 would have been spent on this man's salary. There will of course be other incidental expenses but they will be small by comparison with the salary item. Now, it is not unknown for a design to be completely scrapped because it has been found too heavy when the prototype has been made. If this only happens on an average once every 30 years, then this one man may save easily £100,000 in scrapped tools. Quite apart from speculation on that score, the weight estimator will reduce prices to the extent of the 3s. 10d. per lb plus purchase tax on every car produced. In addition, reduced weight means better performance and lower fuel consumption.

Commercial vehicle operators have stated that every pound of weight saved is worth about 2s. 9d. to them from the point of view of increased carrying capacity. The value of this kind of improvement to the manufacturer as a selling point, and also its prestige value, is very difficult to estimate. There does however seem to be a very strong case for the employment of a weight specialist. The disadvantage is that all drawings must pass through his hands and be signed by him to certify that he has checked them, and made any necessary recommendations concerning them to the appropriate authority. This process will be time consuming, yet a little extra time spent at the design stage will be reflected in gains over the whole output for several years.

It is a comparatively simple matter to calculate the weight of components from drawings. Weights of steel and aluminium sheet per square foot are widely published, and the weight of castings, and machined or forged fittings can be estimated from their volume.

Analysis sheets

In order that an accurate check can be made on weights, it is necessary that records should be kept for each vehicle in the form of weight analysis sheets. These would be headed as shown in the specimen sheet. Upon receipt of a drawing, all the columns 1 to 7, except 4, are to be completed, x, y and z being the distance, in each of the three planes respectively, of the centre of gravity of the component from an arbitrary datum point ; x being the distance forward from the extreme rear of the vehicle, y the distance up from the ground, and z that from one side of the vehicle. The mass moments of inertia I_{xx}, I_{yy} and I_{zz} about the centre of gravity are then calculated and the drawing passed on after any necessary recommendations have been made with a view to reducing the weight of the component. The remaining columns are completed when it is convenient to do so.

At the foot of each page the totals of columns 3 and 8 to 16 inclusive are recorded, and then when the record is complete the totals are brought forward to the end. The horizontal distance of the vehicle's centre of gravity from the datum is then given by $\frac{\Sigma Wx}{W} = \bar{x}$, and its vertical and lateral position by $\frac{\Sigma Wy}{W} = \bar{y}$ and $\frac{\Sigma Wz}{W} = \bar{z}$ respectively. From this a simple calculation of moments will give the front/rear weight distribution. Using the theorem of parallel axes, the pitching, rolling and yawing moments of inertia can then be calculated respectively from :

$$\Sigma W(x^2 + y^2) + \Sigma Izz - W(\bar{x}^2 + \bar{y}^2)$$

$$\Sigma W(y^2 + z^2) + \Sigma Ixx - W(y^2 + \bar{z}^2)$$

$$\Sigma W(x^2 + z^2) + \Sigma Iyy - W(\bar{x}^2 + \bar{z}^2)$$

These moments of inertia are useful for ride and stability criteria, and for this purpose it may be necessary to differentiate between sprung, unsprung and total mass.

When a prototype has been made the estimated moments of inertia can be checked for accuracy by the three-wire method, and of course the centre of gravity position can be checked by measuring the individual wheel reactions. The vertical position of the centre of gravity can be measured by using a spring balance with a horizontal tie to tilt the car sideways on to two wheels. Taking moments about the line of contact of the two wheels with the floor, the actual centre of gravity position can be found. Each individual component should be checked on a weighing machine before assembly, and the true weight recorded in column 4 for comparison with column 3. As experience is gained in this work, the discrepancy between columns 3 and 4 will become less and less. A certain amount of skill and experience is necessary to estimate areas of panels curved in three dimensions.

The most important figures are the weight, fore and aft, and vertical positions of the centre of gravity. It is possible therefore to reduce the work necessary by restricting records to these three quantities. This involves filling in the first nine columns only of the weight analysis sheet, eliminating the most tedious of the calculations, namely those concerned with the moments of inertia. The value of weight distribution estimation lies in the fact that corrective measures can be taken to alter the undesirable centre of gravity position, or to reduce weight in the design stage when alterations can of course be made with a minimum of wasted time and expense.

If savings in weight are to be made, it is necessary to know exactly how the weight is disposed among the components of the car, and for this purpose it is useful to make a weight sheet summary of each type of vehicle, showing how the weight is split up among the major components such as engine, transmission, chassis, body and trim. Given this overall view of the situation, it is then possible for the chief designer to decide exactly where economies can be most effectively made, and in this way the best possible results can be obtained.

Under this system of weight control, the designer will be in a much better position than hitherto to specify with precision how the weight will be distributed on the vehicle. Further, if records of specific weight are kept, a target weight can be easily obtained at the very early stages of the design by multiplying a specific weight obtained from curves similar to those illustrated in this article, by the volume enclosed by the overall dimensions. It is also possible to obtain a direct comparison between cars of different makes, and to determine very definitely just how far any particular policy of refinement at the expense of weight should be carried.

MODEL:									DATE:					
1 Drg No.	2 Component	3 Estd. Weight	4 Actual Weight	5 x	6 y	7 z	8 Wx	9 Wy	10 Wz	11 Wx^2	12 Wy^2	13 Wz^2	14 I_{xx}	15 I_{yy}	16 I_{zz}

Weight sheet headings

CYLINDER HEAD MACHINING

Production Methods for Ford Zephyr and Consul Engines

AS a result of close co-operation between the design engineers and the production engineers, Ford Motor Co. Ltd. are able to machine the cylinder heads for both the six-cylinder Zephyr and the four-cylinder Consul engine on the same machine line without any change of work fixtures or tooling. This is a factor of considerable economic importance since it not only reduces idle time but it also allows capital charges for machines and fixtures to be spread over a much greater number of units than would be possible if two machine lines were necessary.

With only one or two exceptions, the machines have been supplied by James Archdale and Co. Ltd. and Cincinnati Milling Machines Ltd. In accordance with modern machining practice for quantity production, considerable use is made of transfer machines. This, of course, entails the use of open end work fixtures to allow the work to travel in a straight line into the fixture at one end and out at the other. On only two machines is it necessary to move the casting at right angles to the line of travel.

After the casting is received from the foundry, the first machining operation is to mill location pads to give both height and transverse location at the second operation. For this operation a special travelling head Cincinnati vertical milling machine is used, see Fig. 1. It is essential that these pads are machined in accurate relationship to the roofs of the combustion chambers. There is no machining on the contoured form of the combustion chamber, but in the finished head the volume of the chamber must lie within specified limits.

Location for milling the pads is taken from three points. On the four-cylinder head these are, from the roofs of Nos. 1 and 4 combustion chambers in line with the spark plug holes and at 1·00 in from the centre line of the cylinder bores and at one point on the joint face centrally between Nos. 2 and 3 cylinder bores and at 0·38 in from the manifold face. It is worth noting that on the rough casting a tolerance of only 0·005 in is allowed on the relationship between the combustion chamber roof and the joint face.

The operator pushes the casting into the machine along transfer rails. When the casting is in position, the length of transfer rails on which it is resting is dropped to allow the casting to contact the three locating points in the work fixture. Until these contacts are made the machine is inoperable. As soon as the casting is in position a fully automatic work cycle is initiated by means of a press button. To begin the cycle the hydraulic clamps are applied and as soon as the clamping is effected the machine head starts rapid traverse from right to left at 117 in per minute. At the appropriate point in the travel the feed rate, 23·7 in per minute is brought into action. The surfaces to be machined are short and to shorten the cycle accelerated traverse is automatically brought into action between one pad and the next. At the end of the working stroke the clamps are automatically released. Actuation of the clamping fixture is controlled from the spindle carrier movement and until the clamps are released and the casting is raised by the transfer bar until it is clear of the pegs locating in the combustion chambers, the return stroke at rapid traverse cannot begin. On the return stroke, a pawl device mounted on the travelling head contacts the end of the casting to carry the work to the right until it is clear of the work fixture and ready for transfer to the next operation.

For milling the location pads a Galtona shell end milling cutter with inserted carbide tips is employed. The cutter is 6 in diameter and has 12 teeth. Originally it was planned to run the cutter at 180 r.p.m. to give a cutting speed of 282 ft per minute with a feed of 23·7 in per minute. It is however, now being run at a slightly higher speed that gives a cutting speed of 315 ft per minute with 20 in per minute feed. The depth of cut is approximately $\frac{5}{32}$ in and the output is 43 cylinder heads per hour. A 5 h.p. motor is used for the main drive on this machine.

Another special travelling head Cincinnati vertical milling machine is used for the second operation. This is a two-spindle, two-station transfer machine, of the type shown in Fig. 2, on which the joint face is rough machined at the first station and finish machined at the second. With the joint face uppermost, the casting is pushed manually into approximate position at the first station. A fully-automatic work cycle is then initiated for both stations. To begin this cycle, plungers actuated by sequential valving at the back of the machine advance and contact in

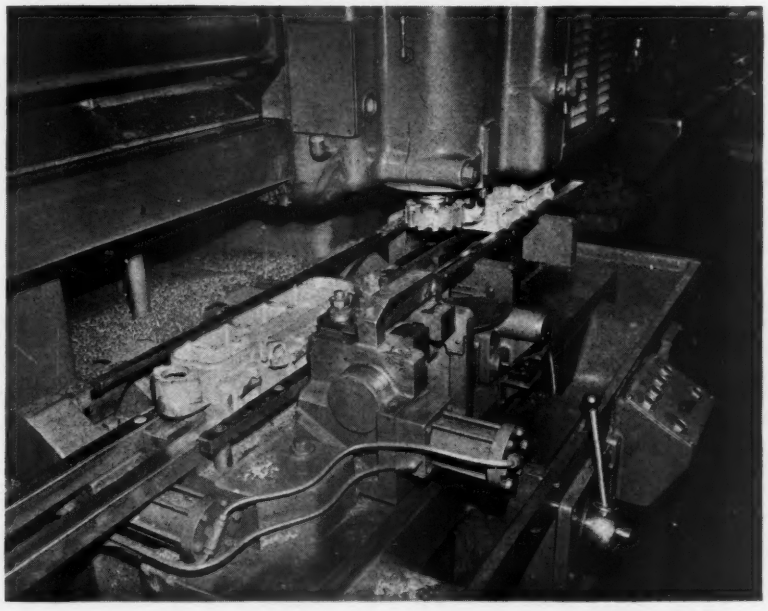


Fig. 1. Special Cincinnati travelling head milling machine for milling location pads.

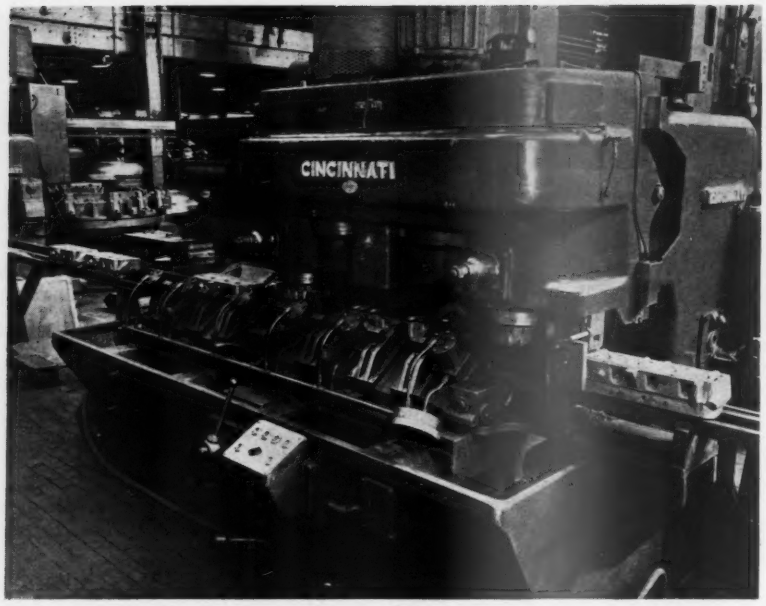


Fig. 2. Special duplex travelling head milling machine for roughing and finishing the joint face. The work is automatically transferred from the first station to the second and is also ejected automatically.

the exhaust pipe seating to push the casting back against a longitudinal stop. As soon as the casting is in contact with the stop, the clamps are automatically applied. When the clamping is effected the head assembly begins to traverse from right to left, starting with fast approach at 193 in per minute and then changing to a feed rate of 25 in per minute.

At the end of the working stroke the clamps are automatically released and the plungers locating in the exhaust pipe seating are withdrawn before the return stroke starts. Transfer from the roughing station to the finishing station and from the finishing station clear of the machine is effected by means of pawl devices mounted on the travelling head. One of these devices is mounted between the machine spindles. On the return stroke, this pawl contacts with a combustion chamber wall to carry the casting forward to the next station. The other pawl is mounted at the extreme right of the travelling head. It contacts the end of the casting at the finish milling station.

On this machine a 25 h.p. motor drives the roughing spindle and a 5 h.p. motor the finishing spindle. A Galtona 8 in diameter face mill with 16 teeth with inserted carbide tips is mounted on each spindle. The roughing cutter runs at 134 r.p.m. to give a cutting speed of 270 ft per minute and the finishing cutter at 143 r.p.m. for a cutting speed of 302 ft per minute. The approximate depths of cuts are $\frac{1}{8}$ in for roughing and 0·020 in for finishing. It is worth noting that on this machine, and indeed on all the milling machines in the line, particular attention has been paid to the question of swarf disposal. Actually, the work fixture has been designed to have as few swarf traps as possible. It is mounted on a trough-shaped bed so that practically all the swarf falls through the fixture down the trough and thence through a chute to an underground swarf conveyor.

At the third operation the manifold face is machined on a special travelling head Cincinnati horizontal milling machine, see Fig. 3. As with the other milling machines the location and clamping arrangements are so interlocked with the movement of the travelling head that traverse for the working stroke cannot start until the work is first properly located and then properly clamped. Similarly, the return stroke cannot be made until the clamps are released.

This machine has a $7\frac{1}{2}$ h.p. motor to drive a spindle carrying a Galtona shell end mill of 6 in diameter and having 12 teeth. The cutter runs at 180 r.p.m. for a cutting speed of 282 ft per minute. As the tool is not cutting over a continuous face, but the distances between the surfaces to be milled are short, the exclusive Cincinnati cam-operated variable feed mechanism is embodied in the machine. Where relatively widely spaced surfaces are to be milled the normal practice is to use rapid traverse between the faces, but where, as at this operation, the distances are relatively short, such a course is not possible. The Cincinnati cam-operated variable feed allows the best possible rate to be used. Actually, the cutting feed rate is 21 in per minute, and the average $25\frac{1}{2}$ in per minute, while rapid

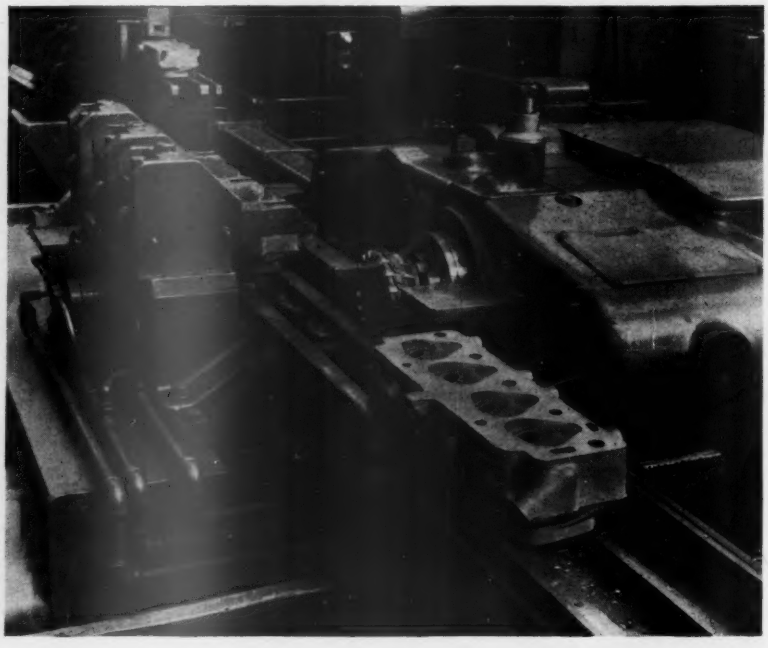


Fig. 3. A Cincinnati travelling head machine with auto ejector for milling the manifold face.

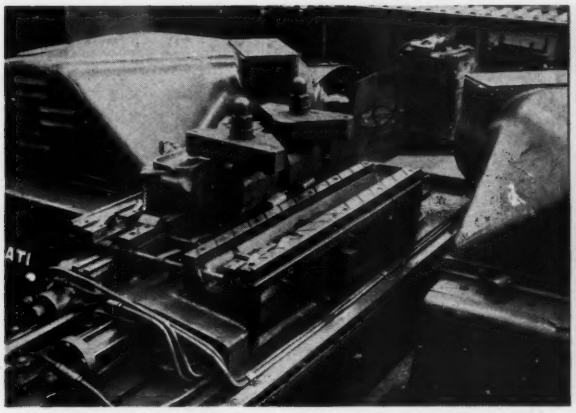


Fig. 4. The loading end of a special machine for milling the exhaust pipe seating.

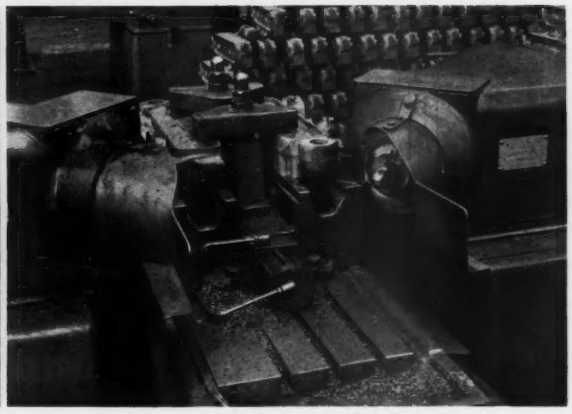


Fig. 5. The work heads of the machine shown in Fig. 4. Two castings are machined simultaneously.

approach and return is at 196 in per minute. The output from this machine is 58 cylinder heads per hour.

Between the third and fourth operations the casting is turned through 180 deg to bring the cover face uppermost ready for rough and finish milling. Location is taken from the joint and manifold faces. The machine used, a special two-spindle, two-station Cincinnati travelling head milling machine, is of the same type as that illustrated in Fig. 2. Two 6 in diameter 12 teeth Galtona shell end mills are used. A 15 h.p. motor drives the roughing spindle at 165 r.p.m. to give a cutting speed of 260 ft per minute, and a 5 h.p. motor drives the finishing spindle at 192 r.p.m. to give a cutting speed of 300 ft per minute. Feed is at the rate of 29·7 in per minute. The approximate depths of cuts are $\frac{1}{8}$ in for roughing and 0·020 in for finishing.

A special Cincinnati 4-48 duplex ram type low bed Hydromatic machine, Figs. 4 and 5, is used to complete the milling operations on the cylinder head. It is fitted with cam-operated variable feed and automatic retraction. On this machine the form of the radial exhaust pipe seating is milled. Special convex end mills are used, and the spindles of the machine are set at 45 deg so that the special cutters do not cut on the nose. These cutters are 0·880 in radius and have four brazed on carbide tips. Each head is driven by a 5 h.p. motor. The spindle speed is 650 r.p.m. to give a cutting speed of 300 ft per minute. Originally an average feed of $12\frac{3}{4}$ in per minute was employed to give an output of 59 four-cylinder heads of 45 six-cylinder heads per hour. Two heads are produced at each machine cycle. It has now been found possible to use an average feed of 19 in per minute.

In establishing the cutting conditions a feed per tooth of 0·012 in was adopted as the basis for the milling operations on the joint, cover and manifold faces. This is higher than usual practice for cast iron with 179–200 Brinell hardness. Experience has shown that at this relatively high rate of feed, the quality of surface finish is very good and the tool life is adequate. These results are obtained because all the factors that influence them have been very carefully considered. For example, every machine has ample power for the work it has to do, the cutters are all mounted with the absolute minimum of overhang, and the clamping is such as to hold the work absolutely rigid even under the heaviest cuts.

Fig. 6. A seven-station Archdale transfer machine from the unloading end.

Drilling and reaming operations

The milling operations provide two accurate height location surfaces, the joint and cover faces, and one accurate transverse location surface, the manifold face. It is however necessary to have an accuration longitudinal location point for the operations that follow. This is provided at the next operation which is carried out on a Pollard two-spindle drill. On this machine a cored water hole in the joint face is reamed. For this operation longitudinal location is taken from the walls of Nos. 1 and 4 combustion chambers. On both the six-cylinder and four-cylinder heads, the reamed location hole is situated between the third and fourth combustion chambers. The second spindle on this Pollard machine

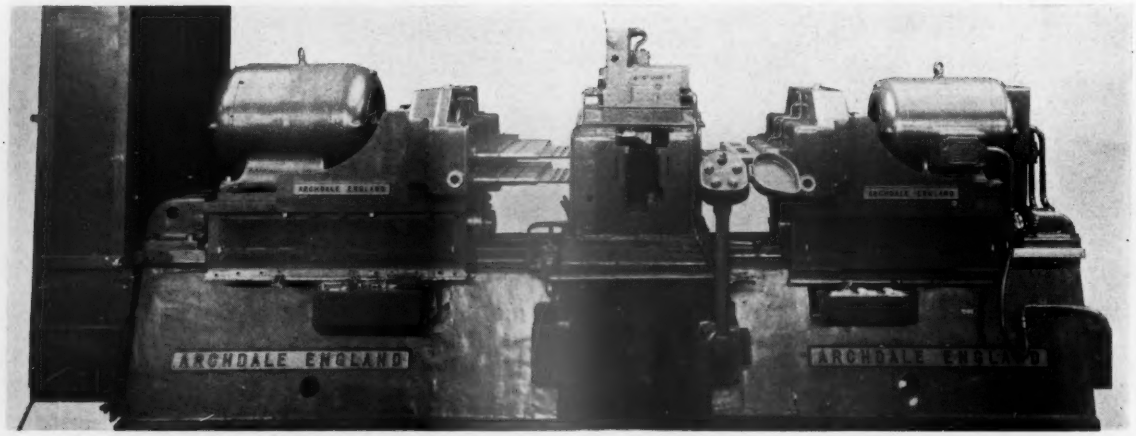


Fig. 7. The first station of the seven-station transfer machine. The left hand head has reciprocating feed for deep hole drilling.

is used for drilling a water hole that cannot be cored in the foundry.

Except that there are more holes to drill, ream and tap in the six-cylinder head than in the four-cylinder head, the operations that follow are exactly similar for each type of head. These notes deal only with the sequence for six-cylinder heads. From the Pollard machine the casting is transferred to a seven-station transfer machine, see Fig. 6. The actual machines are all manufactured by James Archdale and Co. Ltd., but the transfer mechanism is of Ford design. The stations are sufficiently widely spaced to allow easy access to the tools. As a result a fairly long transfer movement would be required if the casting were to be moved direct from one station to the next. To avoid this long movement and to simplify the transfer mechanism, there is an idle station between the successive working stations so that only a relatively short movement is made by the transfer bar. Provision is made for individual operation at each station, if this should be necessary.

For the operations carried out on this seven-station machine the casting has the manifold face resting on the transfer bar. An Archdale horizontal duplex multi-spindle machine is used at the first station. From the left hand head 17 bolt holes are drilled through to 0·465/0·475 in diameter. The spindle speed is 489 r.p.m. and the feed per revolution 0·0085 in. There are 11 spindles in the right hand head for drilling six tapping holes in the rocker pads, two in the rocker cover and two for the water outlet. The holes in the rocker pads are 0·368 diameter and the drills run at 611 r.p.m. For the other five holes drilled from the right hand head the diameter is 0·257 in and the spindle speed is 865 r.p.m. A feed of 0·005 in per revolution is employed. Incidentally, the left hand head has reciprocating feed as the holes to be drilled are of considerable depth and it is advisable to withdraw the drill for the purpose of clearing swarf. This machine is shown in Fig. 7.

At the second station the left hand head is set at an angle of 8 deg to the horizontal for drilling 12 push rod holes to 0·557/0·567 in diameter. For this the drills run at 407 r.p.m. with a feed of 0·011 in per revolution. The right-hand head carries 19 tools, 17 for spotfacing the bolt bosses and 2 for countersinking the water outlet tapping holes. Spotfacing is effected at 229 r.p.m. with a feed of 0·004 in per revolution. This head has an auto-timed dwell at the end of the feed traverse. In fact, every head from which either spotfacing to close limits or counterboring is carried out is designed to give a timed dwell at the end of the feed stroke.

At the idle station between the second and third working stations the component is tilted through 14 deg by means of an air cylinder. This simplifies the construction of the machine since it allows the heads to be mounted on a horizontal bed, see

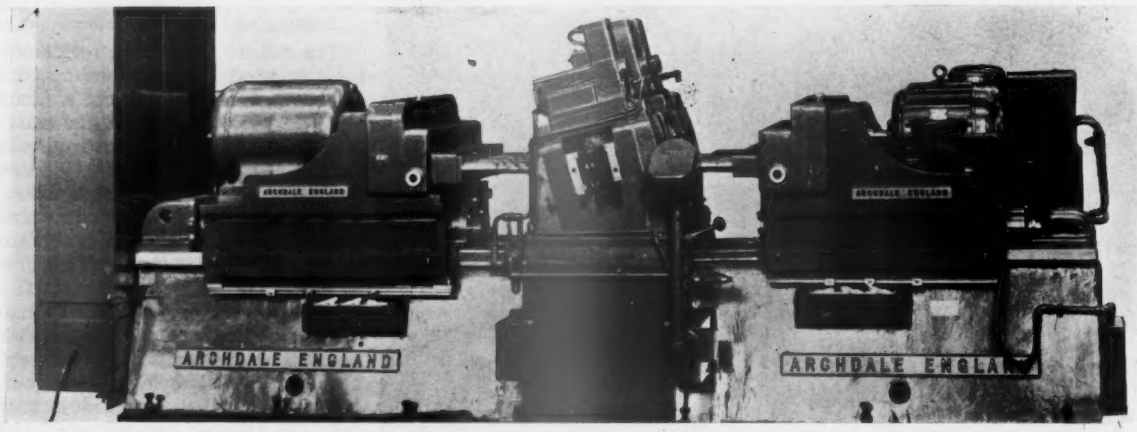


Fig. 8. Archdale machines for the first operations on the valve guide bores and the valve throats. The fixture is at an angle of 14 deg to the spindles.

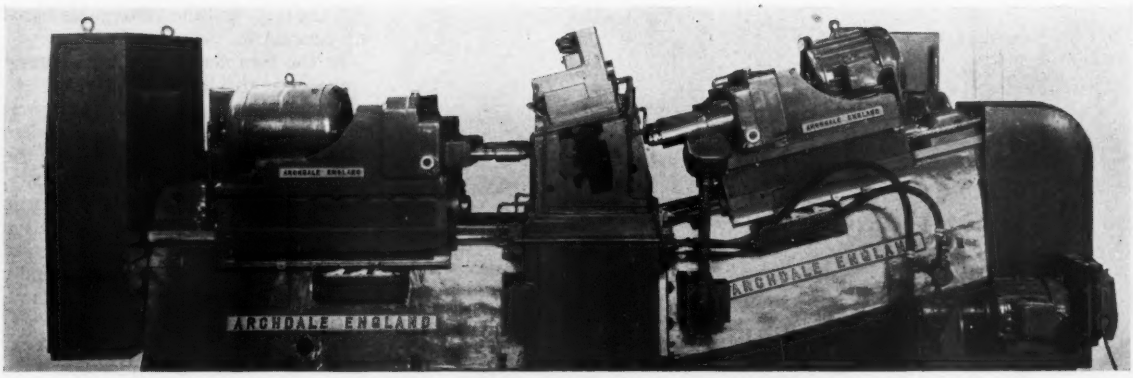


Fig. 9. The final station on the seven-station transfer machine.

Fig. 8, for machining valve throat and valve guide elements which are at an angle of 14 deg to the faces of the casting. From the left hand head of this machine Nos. 1, 3 and 5 exhaust throats and Nos. 2, 4 and 6 inlet throats are rough drilled and formed. The spindles run at 172 r.p.m. and the feed per revolution is 0·022 in. From the right hand head the 12 valve guide holes are drilled to 0·468/0·473 in diameter at 485 r.p.m. and a feed rate of 0·0085 in per revolution. At the fourth working station the remaining exhaust and inlet valve throats are rough drilled and formed from the left hand head, while the left hand head is used to spotface and counterbore Nos. 1, 3 and 5 exhaust and Nos. 2, 4 and 6 inlet valve guide bosses. For the spotfacing and counterboring the spindles run at 167 r.p.m. with a feed of 0·004 in per revolution.

The valve guide holes are semi-finish reamed to 0·480/0·485 in diameter from the left hand head at the fifth station. It will be noticed that the drilling and semi-finish reaming operations on the valve guide holes are carried out from opposite sides. The reamers run at 306 r.p.m. and have 0·014 in feed per revolution. Only a single spindle head is used at the right of this station to drill a 0·432/0·442 in diameter through hole. As this entails deep drilling an Archdale 2A cam unit with a reciprocating cam is employed.

At the sixth working station six form tools are used in the left hand head to finish form the angular throat and semi-finish the parallel portion of Nos. 1, 3 and 5 exhaust ports and Nos. 2, 4 and 6 inlet ports. These tools run at 167 r.p.m. with 0·004 in feed per revolution. The left hand head is used for spotfacing and counterboring Nos. 2, 4 and 6 exhaust and Nos. 1, 3 and 5 inlet valve guide bosses at the same speed and feed. The left hand head at the seventh and last working station of this transfer machine is used to finish form the angular throat and semi-finish the parallel portion of Nos. 2, 4 and 6 exhaust and Nos. 1, 3 and 5 inlet ports. The right hand head carries one tool for boring and counterboring the thermostat hole and nine tools for countersinking the tapping holes in the rocker pads and for the rocker cover. A snout-type boring spindle is used for the boring and counterboring. It runs at 108 r.p.m. and has 0·003 in feed per revolution. This machine is illustrated in Fig. 9.

Two-station transfer machine

On leaving this first transfer machine the casting is turned to rest on the joint face before it goes to the second transfer machine in which location is taken from the joint and manifold faces and the reamed hole in the joint face. This transfer machine has only two stations. At the first station the left hand head is angular and carries tools for drilling six spark plug holes, and two tapping holes and step drilling and counterboring two other holes. The right hand head is used for drilling 34 holes in the intake and exhaust flanges. At the second station only one head is employed. It is used for spotfacing and counterboring the spark plug seats and for counterboring the spanner clearance and for spotfacing a boss. Figs. 10, 11 and 12 show the control panel and the machines. It is scarcely necessary to point out that Figs. 7, 8, 9, 11 and 12 show the machines ready for installation in the line.

At this stage the casting is transferred to an Archdale three-way tapping machine. All the necessary tapping is effected at the one setting on this machine. The average tapping

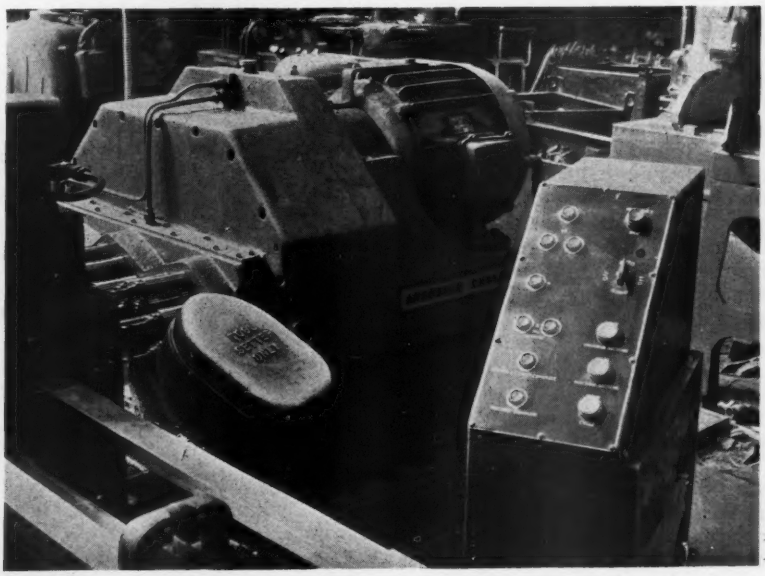


Fig. 10. The control panel and the loading station for a two-station Archdale transfer machine.

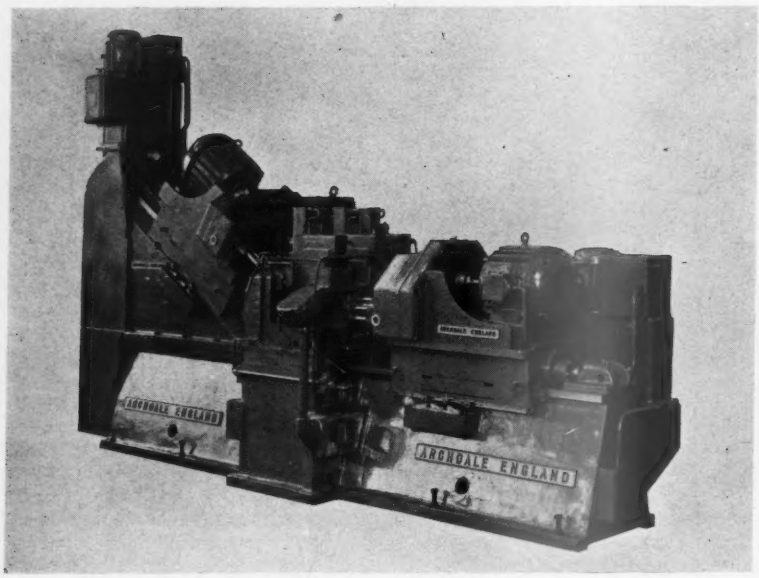


Fig. 11. The set-up for drilling spark plug holes and holes in the exhaust and inlet flanges.

speed is 12 ft per minute. This machine has an independent lead screw for every spindle. There is also independent end adjustment for every spindle so that there is no need to grind the taps in sets to a specified length. In addition, there is a safety tapping chuck on each spindle so that if by any mischance a blind hole is not drilled to the correct depth the tap, after contacting the bottom of the hole will feed up into the chuck as the head continues its feed.

After the tapping operations the valve guide holes are finish reamed on an Archdale machine and then chamfered ready for the insertion of the guides. The guides are then pressed into position and the head then passes through a Curran hot washing machine. On leaving the washing machine, the casting is given an air pressure test in water. This test is carried out in a Ford designed test rig, illustrated. Immediately after the test the casting is again washed. It is then thoroughly blown out with compressed air to remove any swarf that may have accumulated during the machining operations.

Final operations

Two further operations complete the machining on the cylinder head. They are both carried out on Archdale vertical hydraulic drilling machines. At the first of these operations reamers are passed down the valve guides to correct the very slight closing in that occurs when the guides are pressed into the head. At the same setting a light skimming cut is taken on the edges of the inlet and exhaust valve throats. The final operation is to finish cut the inlet and exhaust valve seats. For this operation the tools are carried in floating drive holders and have long pilots which register in the valve guide bores to ensure that the seats and the valve guide bores are concentric.

At the completion of the machining operations, the head is again washed and then blown out with compressed air before it is given a complete inspection prior to transfer to the engine assembly section. Incidentally, although reference has not been made to inspection during the operation sequence, there are several inspection stations in the line. They are generally immediately after one of the more critical operations.

This machine line is an excellent example of balanced production. In general there is little difference in the cycle times for all the individual operations. In fact, the balanced cycle time is one of the outstanding features. Reference should also be made to the manner in which the line has been laid out to obtain and maintain high production rates without causing any marked degree of operational fatigue. Another noteworthy feature is the clean design of the work-holding fixtures. These, and particularly on the milling operations, are of extremely open design and are consequently almost completely free from swarf traps. Nevertheless, they are also designed to hold the work absolutely rigidly so that a close degree of accuracy is maintained.

Fig. 12. The machine used at the second station of the two-station transfer machine.

ROLL ANGLES

The Calculation of Wheel Loads and Angular Movement on Curves

By Prof. Dr. Ing. Robert Eberan v. Eberhorst*

THE rolling of a vehicle and the associated changes in wheel loads are of importance in establishing the behaviour of an automobile when negotiating curves or when subject to sudden changes of direction; the loading and orientation of individual wheels determines the forces available to keep the vehicle on its desired course, while the roll-frequency determines, among other things, the reaction time under a suddenly imposed change of direction. A method of calculating the roll-angle and the wheel loads should therefore be of value in the design of a vehicle or in making comparisons between different methods of suspension.

A universally applicable system of calculation has been developed, enabling the roll angle and wheel loads to be calculated for any vehicle whose dimensions and weights are known. After the development of this system follows a comparison of different suspension arrangements in respect of roll angle and wheel load distribution, and the influence of an anti-roll bar is also considered. In an Appendix actual calculations are made for three cars of average dimensions fitted respectively with independent front and beam rear suspension, with independent suspension at both front and rear, or with independent front suspension and a swing-axle at the rear. All three examples are, in addition, considered both with and without an anti-roll bar on the front suspension.

The symbols used in the formulae are given in the following list; it will be noted that dimensions are given in inches, forces in pounds and angles in radians.

a	=	distance from c.g. of sprung masses to front axle	in
a_1	=	distance from centre line of chassis to outer pivot point (only correct for very small roll angles)	in
a_2	=	distance from centre line of chassis to spring base of rear suspensions	in
b	=	distance from c.g. of sprung masses to rear axle	in
b_1	=	distance from the instantaneous centre to the outer suspension pivot point	in
b_3	=	width of spring blade	in
c	=	wheel base	in
c_1	=	distance from instantaneous centre to tyre contact point	in
d	=	diameter of spring wire	in
d_1	=	distance from roll centre to tyre contact point	in
d_3	=	dia. of coil spring	in
e	=	length of leverage of bar	in
f	=	length of swing axle	in
h	=	height of c.g. of sprung masses above road	in
h_1	=	thickness of one leaf of spring	in
l	=	effective spring length	in
m	=	height of roll axis above road, front	in
n	=	height of roll axis above road, rear	in
p	=	distance between anti-roll bar bearings	in
p_1	=	distance between anti-roll bar links	in
r	=	rolling radius of tyre	in
r_1	=	radius of curve	ft
s	=	height of c.g. above road surface	in
t_1	=	front track	in
t_2	=	rear track	in
w	=	distance from centre line of chassis to inner pivot point	in
C, C_1, C_2, C_3, C_4	=	spring rates	lb/in
E	=	modulus of elasticity	lb/in^2
F	=	front axle weight (static)	lb
F_1, F_2	=	front wheel loads (car rolling)	lb
G	=	modulus of rigidity	lb/in^2
M_1, M_2, M_3, M_4	=	moments of forces acting on the sprung masses	lb-in
P	=	reaction force of anti-roll bar in chassis	lb
P_1	=	force in anti-roll bar link	lb
R	=	rear axle weight (static)	lb
R_1, R_2	=	rear wheel loads (car rolling)	lb
T	=	total weight of vehicle	lb
W	=	weight of sprung masses	lb
W_1	=	weight of front unsprung masses	lb
W_2	=	weight of rear unsprung masses	lb
ΔF	=	dynamic increase (or decrease) in front wheel load	lb
ΔR	=	dynamic increase (or decrease) in rear wheel load	lb
α	=	angular movement of wheel about instantaneous centre	radn
γ	=	angular movement of suspension arm	radn
μ	=	$\frac{\Theta}{g} = \frac{\text{Centripetal acceleration in the curve}}{\text{Acceleration of gravity}}$	
Ψ	=	roll angle of the sprung masses (angle between the vertical axis of the vehicle and the perpendicular to the road surface)	radn
n_F	=	natural frequency of sprung mass, front	
n_R	=	natural frequency of sprung mass, rear	
n_1	=	total number of leaves	
n_2	=	number of main leaves	
$\frac{u}{v}$	=	leverage ratio of suspension arm	
V	=	speed in ft/sec	
M_0	=	roll centre	
M	=	instantaneous centre	
i	=	number of working coils	

*Chief Engineer, David Brown Tractors (Engineering) Ltd., Automobile Division.

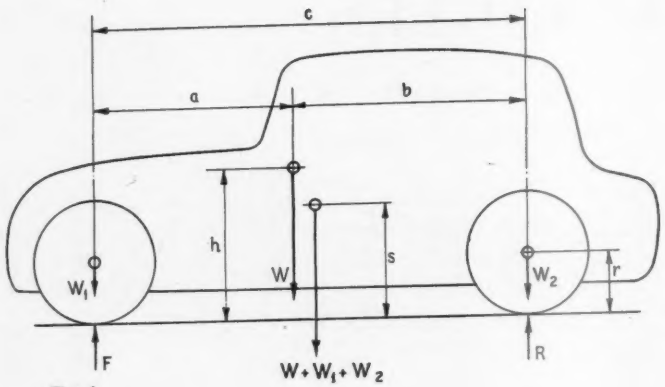


Fig. 1.

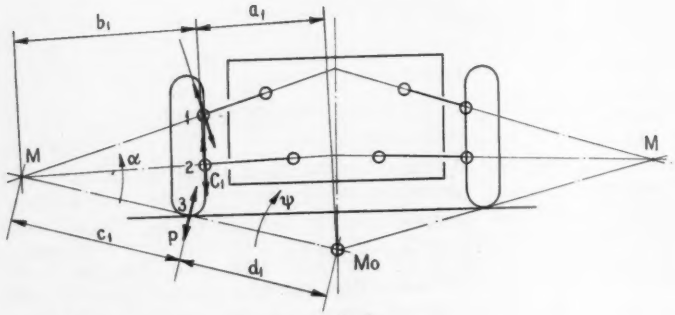


Fig. 2.

$$M_1 = 2\,\frac{b_1^2\,d_1^2}{c_1^2}\,C_1\,\psi$$

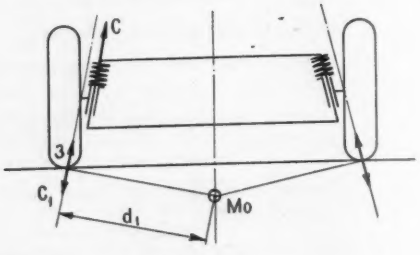


Fig. 3.

$$C = C_1$$
$$M_1 = 2d_1^2\,C_1\,\psi$$

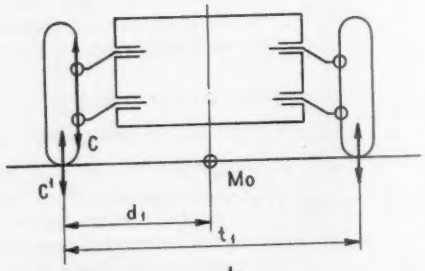


Fig. 4.

$$d_1 = \frac{t_1}{2}$$
$$C = C_1$$
$$M_1 = \frac{t_1^2}{2}\,C_1\,\psi$$

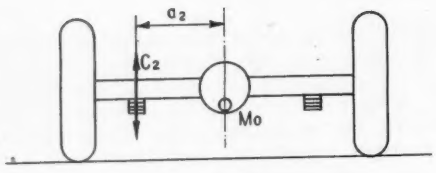


Fig. 5.

$$M_2 = 2\,C_2\,a_2^2\,\psi$$

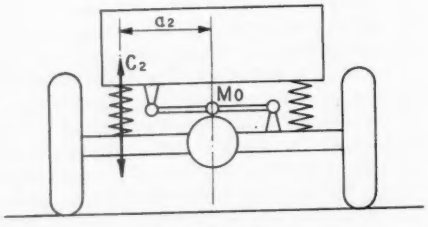


Fig. 6.

$$M_2 = 2\,C_2\,a_2^2\,\psi$$

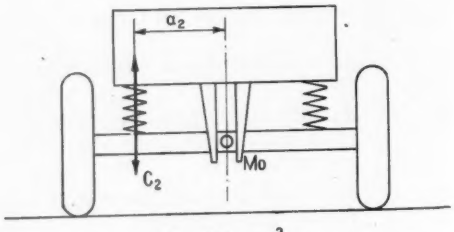


Fig. 7.

$$M_2 = 2\,C_2\,a_2^2\,\psi$$

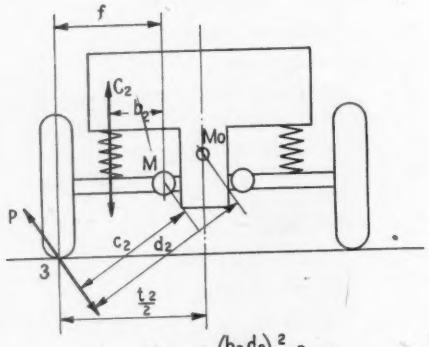


Fig. 8.

$$M_2 = 2\left(\frac{b_2 d_2}{c_2}\right)^2 C_2\,\psi$$
$$\frac{1}{2}\left(\frac{b_2 t_2}{f}\right)^2 C_2\,\psi$$

Position of centre of gravity

The horizontal location of the centre of gravity, see Fig. 1, is obtained by weighing the front and rear axles. Its height is most easily determined by inclining the vehicle laterally until the over-turning point is reached. When doing this the wheels must be blocked in their normal running position and the tyres blown up so hard that their lateral deformation is negligible. The c.g. of the unsprung masses, whose weights are W_1 and W_2, should be taken as lying at the wheel centres. For the determination of the position of the c.g. of the sprung masses, in so far as it concerns the calculation of the rolling tendency, the following relations hold good:

$$W = T - W_1 - W_2$$

$$a = c.\frac{R - W_2}{W}$$

$$b = c.\frac{F - W_1}{W}$$

$$h = \frac{T.s - (W_1 + W_2)\,r}{W}$$

Roll centres

To determine the roll centre with independently sprung wheels the instantaneous centre of each wheel in its movement in relation to the chassis must first be ascertained. For a link mounting, as in Fig. 2, the wheel attachment points 1 and 2 move at right angles to the links and the instantaneous centre M of the wheel is therefore at the intersection point of the produced axes of the links. The movement of the tyre contact point 3 about the instantaneous centre is then at right angles to the connecting line M—3 and the roll-centre of the chassis must lie on M—3 produced at M_0 if the wheels be assumed to remain on the road and that the chassis undergoes the rotary movement. Symmetry being assumed, the roll-centre must lie on the central plane of the chassis.

For a wheel mounting according to Fig. 2, the roll-centre lies below the road surface but can be brought above the road plane by suitable inclination of the cross linkage.

It is not the purpose of this investigation to discuss the layout of this linkage from the point of view of minimizing camber-angle changes and track variations, but the various possible positions of the roll-centre for different types of independent suspension will be examined. With horizontal, parallel links which can, be it noted, have any desired different lengths, the instantaneous centre of the wheel movement in relation to the chassis lies at infinity. The line M—3 is therefore horizontal and the roll-centre lies on the road surface. This is the condition approximately obtaining in most modern front suspension arrangements.

Parallel guidance of the wheels, as in the sleeve mounting of the Lancia front wheels or in the double-crank suspension of Porsche, can be considered as link-mounting with the members of infinite length. The roll-centre of such arrangements can be found by drawing a line through the tyre contact point 3 perpendicular to the axis of the guide system and producing it to meet the central plane of the chassis, as shown in Figs. 3 and 4. Leaf cross-springs can be treated as links of about 0·75 to 0·77 the length of the actual springs.

In the case of a beam or De Dion axle carried on leaf springs the roll centre M_0 can be assumed to lie at the same height as the spring anchorages, see Fig. 5. The exact height of the roll-centre is, however, dependent on the degree of lateral elasticity of the spring mounting and cannot therefore be exactly forecast.

Where lateral location is afforded by a cross-link, as in Fig. 6, a ball socket sliding in a vertical guide, Fig. 7, or by a triangular link between the frame and the axle, the roll-centre lies at the height of the attachment of the link to the chassis. With wheel mounting systems according to Figs. 2, 6 and 8 the height of the roll-centre above the road varies with the spring deflection and depends therefore on the loading of the vehicle.

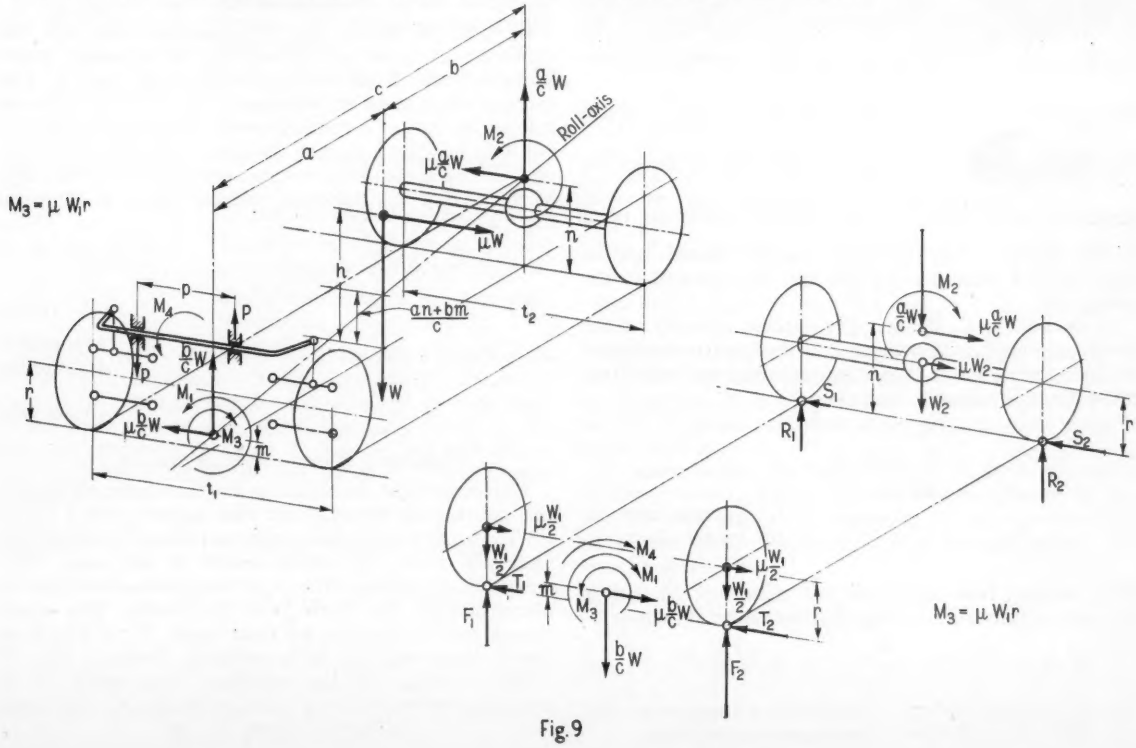
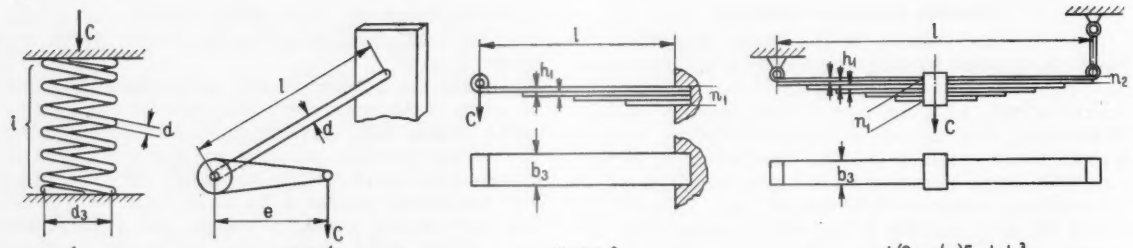


Fig. 9



$C = \frac{G d^4}{8 l d_3^3}$ lb/in $G = 11{\cdot}5 \times 10^6$ lb/in²

$C = \frac{G \pi d^4}{32 e^2 l}$ lb/in $G = 11{\cdot}5 \times 10^6$ lb/in²

$C = \frac{E n_1 b_3 h_1^3}{6 l^3}$ lb/in $E = 29 \times 10^6$ lb/in²

$C = \frac{4 (2 + n_2/n_1) E n_1 b_3 h_1^3}{3 l^3}$ lb/in $E = 29 \times 10^6$ lb/in²

Fig. 10

In consequence of the generally different heights of front as against rear roll-centres, m and n, the roll-axis of the sprung masses is generally not horizontal and parallel to the road surface, as Fig. 9 makes clear. The roll axis of modern private cars falls towards the front, since standard practice embodies independently sprung front wheels with a roll-centre close to the road surface and a beam—or swing axle—at the rear, with a roll-centre roughly at wheel centres. The effective moment-arm of centrifugal force on the chassis μW is thereby reduced to $h - \left(\frac{an + bm}{c}\right)$.

For swing axles, the roll centre lies on the intersection with the central plane of a line passing through the tyre contact point 3 and through the hinge point of the axle arm M, as indicated in Fig. 8.

Spring forces and moments

Spring rates which can be calculated from the spring dimensions for the most commonly used systems—the coil spring, the torsion bar, the quarter-elliptic and the half-elliptic leaf spring—are given in Fig. 10. When an intermediate lever is used, as shown in Fig. 11, the rate of the spring C must be multiplied by the square of the leverage ratio $\left(\frac{u}{v}\right)^2$ to obtain the effective rate C_4 at the lever end. It will be understood that the leverage $\frac{u}{v}$ reduces the forces in the ratio $\frac{u}{v}$ but at the same time increases the movement in the ratio $\frac{v}{u}$. Where rubber bushes are fitted to the linkage, their restoring couples should not be neglected but measured by test and incorporated in the spring rates.

If, in Fig. 2, C_1 denotes the specific elasticity at the lower suspension arm at point 2, α the angular movement of the wheel about its instantaneous centre M, and Ψ the roll-angle of the chassis, then :—

$$c_1 . \alpha = d_1 . \Psi$$
$$P . c_1 = b_1 . \alpha . C_1 . b_1$$
$$M_1 = 2P . d_1$$
$$M_1 = 2\left(\frac{b_1 d_1}{c_1}\right)^2 . C_1 . \Psi$$ General case for independent suspensions.

With parallel, horizontal links the roll-centre M_0 lies in the road surface and the instantaneous centre at infinity :—

$$b_1 = c_1 = \infty \, , \; \frac{b_1}{c_1} = 1 \, , \; d_1 = \frac{t_1}{2}$$

$M_1 = \frac{t_1^2}{2} . C_1 . \Psi$ Independent suspensions with horizontal parallel links.

$M_1 = 2 d_1^2 . C_1 . \Psi$ Independent suspensions with sleeve guiding, Fig. 3. Double-crank, Fig. 4

For an angular movement α of a swing axle about its instantaneous centre M the same considerations apply as for the general case of independently sprung wheels, Fig. 2 :—

$$C_2 . \alpha = d_2 . \Psi$$
$$P . c_2 = b_2 . \alpha . C_2 . b_2$$
$$M_2 = 2 . P . d_2$$
$$M_2 = 2\left(\frac{b_2 d_2}{c_2}\right)^2 . C_2 . \Psi$$ Swing-axle, Fig. 8.

Or also :—

$$M_2 = \tfrac{1}{2}\left(\frac{b_2 t_2}{f}\right)^2 . C_2 . \Psi$$

For a beam axle the roll angle Ψ is identical with the displacement angle at the spring attachment point and the restoring couple becomes :—

$M_2 = 2 C_2 . a_2^2 . \Psi$ Beam-axle, Figs. 5, 6 and 7.

Anti-roll bar

In order to reduce the roll-angle, but also with the deliberate intention of influencing the dynamic wheel loads, an anti-roll bar is used, as shown in Fig. 12. The spring rate C_3 of an anti-roll bar is calculated in the same manner as that of a torsion spring according to Fig. 10, but only half the length of the bar is considered for each side. Between the stabilizing moment M and the roll-angle of the chassis the following relations hold, as shown in Fig. 12 :—

$$a_1 \Psi = v . \gamma$$
$$P_1 = C_3 . u . \gamma = \frac{a_1 u}{v} . C_3 . \Psi$$
$M_4 = P . p = P_1 . p_1 = \frac{a_1 p_1 u}{v} . C_3 . \Psi$ Independent suspension.

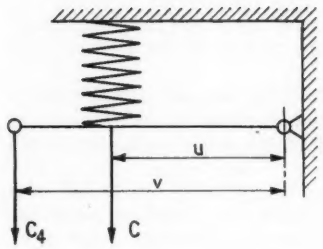
$M_4 = \frac{p_1^2}{2} . C_3 . \Psi$ Beam axle.

Influence of the unsprung masses

The centrifugal force acting on the unsprung masses and the reaction forces at the road surface gives a couple tending to tilt the wheels in the same sense as the centrifugal force μW turns the chassis about its roll axis. With independently sprung wheels the suspension links transfer tilting moment $M_3 = \mu W_1 . r$ to the chassis. The wheels change their camber by the same angle Ψ, by which the chassis leans outwards on a curve, as shown in Fig. 12.

With a swing axle the centrifugal force acting on the unsprung masses exerts a righting couple on the chassis

$$M_3 = -\mu W_2 (n - r)$$



$$C_4 = C\left(\frac{u}{v}\right)^2$$

Fig. 11

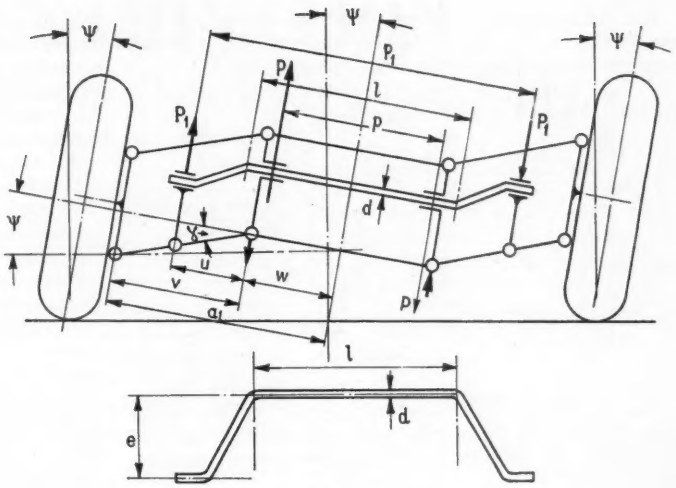


Fig. 12

if the roll-centre M_0, as in Fig. 8, lies higher than the wheel centre. This moment must be included, with the correct sign, in the balance of forces on the chassis.

With a beam axle the centrifugal tilting moment $M_3 = \mu W_2 . r$ of the unsprung masses can naturally have no influence on the inclination of the chassis, but it does affect the dynamic wheel loads (see section: Dynamic wheel loads).

Roll angle of chassis

From the balance of moments about the roll axis, see Fig. 9, upper view, the roll angle for any centripetal acceleration can be calculated as follows :—

$\mu W . \left[h - \frac{an+bm}{c}\right]$	Centrifugal moment on chassis.
$+(\mu . W_1 . r)$	Tilting moment of the unsprung masses with independently sprung front wheels.
$+(\mu . W_2 . r)$	Tilting moment of the unsprung masses with independently sprung rear wheels.
$=M_1$	Spring restoring couple on front axle.
$+M_2$	Spring restoring couple on rear axle.
$+(\mu W_2 [n-r])$	Negative tilting moment of the unsprung masses with a swing axle.
$+(M_4)$	Stabilizing moment of the anti-roll bar.

According to the design of the axles this equation is to be used with or without the applicable terms shown in brackets ( ). As an example, the equation to determine the roll-angle of a vehicle with independent front wheels, front roll centre on the road (m=0), an anti-roll bar in front and a beam axle at the rear, as in Fig. 9 would read as follows :—

$$\mu . W . \left[h - \frac{an}{c}\right] + \mu . W_1 r = M_1 + M_2 + M_4$$

$$= C_1 \Psi \frac{t_1^2}{2} + C_2 \Psi 2a_2^2 + C_3 \Psi \frac{a_1 p_1 u}{v}$$

$$\Psi = \frac{\mu W \left[h - \frac{an}{c}\right] + \mu W_1 r}{C_1 \frac{t_1^2}{2} + C_2 . 2a_2^2 + C_3 \frac{a_1 p_1 u}{v}}$$

The roll angle Ψ, assuming straight-line spring deflection characteristics, is proportional to the centripetal acceleration and increases as the square of the speed, since $\Theta = \mu g = \frac{V^2}{r_1}$ in which V is the speed and r_1 the radius of the curve.

In this method of calculating the chassis roll-angle no account has been taken of the deformation of the tyres nor has the gravity tilting moment of wheels running under camber been allowed for. The centrifugal force acting on the chassis has, moreover, been assumed to be applied at the centre of gravity, whereas in fact the centrifugal moments are distributed over the whole length of the vehicle. The chassis has been assumed to be rigid, a condition which does not obtain in practice.

Allowance for torsional deflection of the frame when obtaining the roll-angle and the wheel loads would require an accurate knowledge of the local stiffness of the frame and would greatly complicate the calculation process without appreciably altering the basic results.

Dynamic wheel loads

The tilting moment of the centrifugal force causes a corresponding addition ΔF and ΔR to the wheels on the outside of the curve, while the wheels on the inner side have their loads reduced by the same amount :—

$$F_2 - F_1 = 2\Delta F \qquad F_2 = \frac{F}{2} + \Delta F \qquad F_1 = \frac{F}{2} - \Delta F$$

$$R_2 - R_1 = 2\Delta R \qquad R_2 = \frac{R}{2} + \Delta R \qquad R_1 = \frac{R}{2} - \Delta R$$

Since the chassis, with its support on four wheels is a "statically indeterminate system," direct calculation of F_1, F_2 and R_1, R_2 is not possible. The roll-angle Ψ must first be calculated and then the restoring couple at each axle worked out.

To make clearer the balance of the loads on the axles, the lower view in Fig. 9 shows a layout of forces on the axles identical with those shown acting on the chassis in the upper view, the forces and moments having the opposite sense :—

$$M_1 + \mu \frac{b}{c} Wm + M_4 = (F_2 - F_1) \frac{t_1}{2}$$

$$\Delta F = \left[\frac{1}{t_1} 2 \left(\frac{b_1 d_1}{c_1}\right)^2 C_1 . \Psi + \mu \frac{b}{c} Wm + \frac{a_1 p_1 u}{v} . C_3 \Psi\right]$$

Independent suspension with anti-roll bar.

The tilting moment on the unsprung masses $\mu W_1 r$ is removed from the frame by the reaction M_3 and has therefore no influence on the wheel loads F_1 and F_2.

$$M_2+\mu\,\frac{a}{c}\,Wn+\mu W_2 r+M_4=(R_2-R_1)\,\frac{t_2}{2}$$

$$\Delta R=\frac{1}{t_2}\left[2a_2{}^2\,C_2\,\Psi+\mu\,\frac{a}{c}\,Wn+\mu W_2 r+\frac{p_1{}^2}{2}.C_3.\Psi\right]$$

Beam axle with anti-roll bar.

$$M_4+M_2+\mu.\frac{a}{c}.W.n+\mu.W_2 n=(R_2-R_1)\,\frac{t_2}{2}$$

$$\Delta R=\frac{1}{t_2}\left[\left(\frac{b_2+t_2}{f}\right)^2.\frac{C_2\Psi}{2}+\mu\frac{a}{c}Wn+\mu W_2 n+\frac{a_1 p_1 u}{v}.C_3.\Psi\right]$$

Swing axle with anti-roll bar.

The above equations for obtaining the dynamic wheel loads ΔF and ΔR are equally applicable when no anti-roll bar is fitted, the last term in the expression merely vanishes since $C_3=0$.

Appendix

Three different chassis layouts, with and without anti-roll bar and of normal dimensions are compared numerically :—

I Independent front suspension, beam axle (De Dion) at rear.

II Independent front and rear.

III Independent front, swing axle rear.

All three vehicles are assumed to have the same suspension periodicities of 75 per min front and 85 per min rear and the spring rates C_1 and C_2 are calculated to correspond. Allowance has been made for difference in unsprung weights consequent in change from De Dion to swing axle and for the consequent change in the position of the centre of gravity of the sprung mass (see figures in table). In all cases the rate of the anti-roll bar on the front suspension has been taken as the same as that of the load-carrying springs $C_3=C_1$.

No attempt has been made to compare actual maker's vehicles, since differences in dimensions, weights, centre of gravity and spring deflections would make true comparisons impossible.

The smallest roll-angle Ψ is given by design III with anti-roll bar ; this vehicle without the anti-roll bar shows the greatest dynamic change of rear wheel loading. Some of the values of wheel loads given in the table are negative, which means that under the assumed value of $\mu=1$ that particular wheel would be lifted from the road. All three designs, when the anti-roll bar is omitted and $\mu=1$, have the characteristic that the rear wheel on the inside of the curve lifts from the road. The anti-roll bar prevents this happening on designs I and II, but not on design III. In design II the fitting of the anti-roll bar transfers the wheel lift to the inside *front*.

The critical value given for μ represents that centrifugal acceleration at which one wheel just begins to lift. It must be observed here that centrifugal accelerations as high as g do not occur in practice, μ reaching a maximum of about 0·8, at which the drift angle of the tyres is well over 10 deg. In spite of the fact that the calculations were carried out on a basis of $\mu=1$ all three designs are in practice quite safe against wheel lift.

By fitting an anti-roll bar with a rate $C_3=C_1$, the dynamic wheel loads can be changed by 30 to 40 per cent and the roll angle reduced by 30 to 37 per cent, which shows that this fitting can have a notable influence on the road behaviour of a vehicle.

T	=	2800 lb		c	=	105 in		n_F	=	75 /min
F	=	1400 lb		t_1, t_2	=	54 in		n_R	=	85 /min
R	=	1400 lb		s	=	24 in				
F : R	=	50 : 50%		r	=	13·5 in				

		I		II		III	
Front Axle		**Independent**		**Independent**		**Independent**	
Back Axle		**Beam**		**Independent**		**Swing Axle**	
Anti-Roll Bar		**without**	**with**	**without**	**with**	**without**	**with**
Unsprung weight, front	W_1 lb	200 lb		200		200	
Unsprung weight, rear	W_2 lb	300 lb		200		200	
Dimensions	a in	50·25 in		52·5		52·5	
,,	b in	54·75 in		52·5		52·5	
,,	h in	26·25 in		25·75		25·75	
,,	m in	0		0		0	
,,	n in	13·50 in		0		15·0	
,,	b_2 in	21·0 in		—		18·3	
,,	f in	—		—		24·3	
Spring Rates	C_1 lb/in	95·6 lb/in		95·6		95·6	
,, ,,	C_2 ,,	112·8 lb/in		123·0		217·0	
,, ,,	C_3 ,,	0	95·6	0	95·6	0	95·6
For μ = 1·0 Roll-angle	Ψ radn	0·2019	0·1273	0·2110	0·1465	0·1449	0·1007
,,	Ψ deg	11·55	7·30	12·1	8·40	8·30	5·77
Dynamic addition load	ΔF lb	521	658	545	758	374	521
,, ,,	ΔR lb	723	585	701	487	870	723
Dynamic wheel load	F_2 lb	1221	1358	1245	1458	1074	1221
,, ,,	F_1 lb	180	43	155	—58	326	180
,, ,,	R_2 lb	1423	1285	1401	1187	1570	1423
,, ,,	R_1 lb	—23	115	—1	213	—170	—23
Critical value of μ		0·970	1·064	0·999	0·924	0·804	0·970

TYRE EXAMINATION

Ultrasonic Equipment for the Detection of Cavity Faults

By P. Hatfield, M.Sc., A.Inst. P.*

A PNEUMATIC tyre is essentially a composite structure assembled, in the larger sizes, from as many as 60 to 80 separate components. The final moulding and vulcanization process unites these components into a uniformly bonded whole. However, it may sometimes happen that there is a bonding fault over a small area, which cannot be detected by the normal examination techniques. Such a fault may be present in a new tyre, or a more likely possibility is for it to occur as a result of prolonged or severe service. It is very desirable to be able to locate any such areas of looseness, particularly when a tyre casing is examined with a view to retreading. Where there is an area of looseness between any two components in a tyre, there is also an air film. The property of ultrasonic waves transmitted under water in being reflected by an air film, has been developed into a valuable technique for disclosing areas of looseness between the component parts in a tyre casing. By this method areas of looseness as small as $\frac{3}{8}$ in $\times \frac{3}{8}$ in can be located.

Properties of ultrasonic waves

Ultrasonic waves are essentially sound waves, longitudinal vibrations of frequencies above the audible limit, that is above 20 kc/s. Their main feature, which is so useful to industry, is their 100 per cent reflection at a solid-air boundary. This property is made use of not only in the rubber industry, but much more widely in the metal industries, to detect internal air cavities in solids. Just as some materials will reflect light rays and other materials will transmit light rays, so some substances will reflect and some transmit ultrasonic waves. For example, if a beam of ultrasonic waves passes from water through rubber, the rubber is accoustically transparent, and most of the ultrasonic energy passes through the rubber without any reflection. However, at a water-metal interface, the reflection coefficient is about 50 to 90 per cent depending upon the metal. At a rubber-air or metal-air interface, the reflection is 100 per cent and it is this fact alone that is the basis of any ultrasonic flaw detection system.

Two other physical properties of ultrasonic waves must be considered in designing a method of detecting internal air cavities in solids. They are the beaming and absorption of the ultrasonic waves. Sound waves are very diffuse and radiate in every direction, but ultrasonic waves are directional. At low ultrasonic frequencies the angle of the main beam is very wide, but at higher ultrasonic frequencies, say about 1 Mc/s, the radiated beam is almost parallel. Another factor that determines the directional pattern is the diameter of the ultrasonic source. The larger the diameter, the more directional the beam becomes. However, there is a practical limit to the diameter of the source. The absorption of the waves in rubber is important, because the absorption of ultrasonic waves in rubber increases rapidly with increasing frequency. The absorption in rubber is relatively high compared with metals and water, and the practical frequency limit for testing tyres is about 500 kc/s. The drop in signal strength of an ultrasonic wave passing through 1 cm of rubber at 500 kc/s frequency can be as high as 95 per cent depending on the rubber. The corresponding drop for water at the same frequency would be negligible.

Ultrasonic waves

There are various methods of producing and detecting ultrasonic waves, but one of the most suitable methods which we use is a vibrating quartz crystal. Quartz belongs to a class of crystals which are piezo-electric. In 1880 the Curies found that when the faces of certain crystals were pressed, electric forces were developed on their faces. The converse effect was discovered a year later by Lippman who found that applying an electric field to a piezo-electric crystal caused it to change in length. If an alternating electric field was applied, then the crystal would vary in length at the same frequency as the applied electric field. With quartz the effect is greatest if the frequency of the applied electric field is the same as the resonant frequency of the quartz.

The resonant frequency of an X-cut quartz is determined by its thickness. A quartz crystal 3 mm thick, resonates at about 1 Mc/s frequency and a quartz 30 mm thick resonates at 100 kc/s. Because quartz is expensive, one would not normally use a crystal 30 mm thick but would use a 3mm quartz and stick thick steel electrodes on to it to make a composite crystal which vibrates at 100 kc/s. The steel-quartz construction is used by us in our ultrasonic applications and can be used either to transmit or receive ultrasonic waves. The crystal is mounted in a brass holder, which is sealed to prevent ingress of water into the

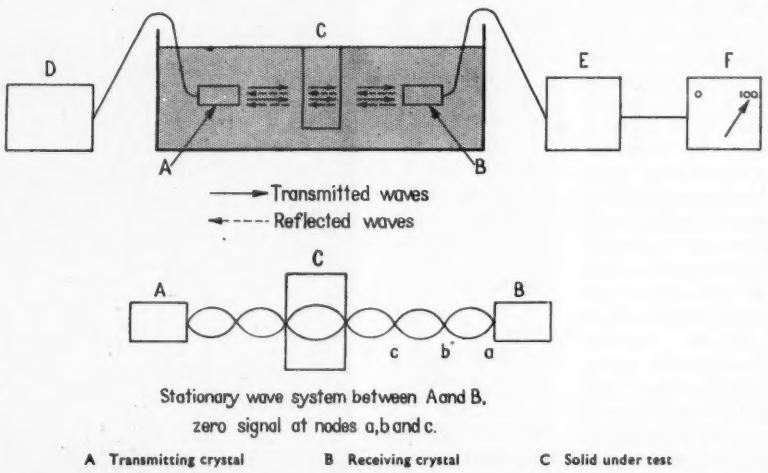


Fig. 1. Typical high-frequency, continuous wave, ultrasonic system for detection of air cavities in solids.

* Physical Research Division, Dunlop Research Centre.

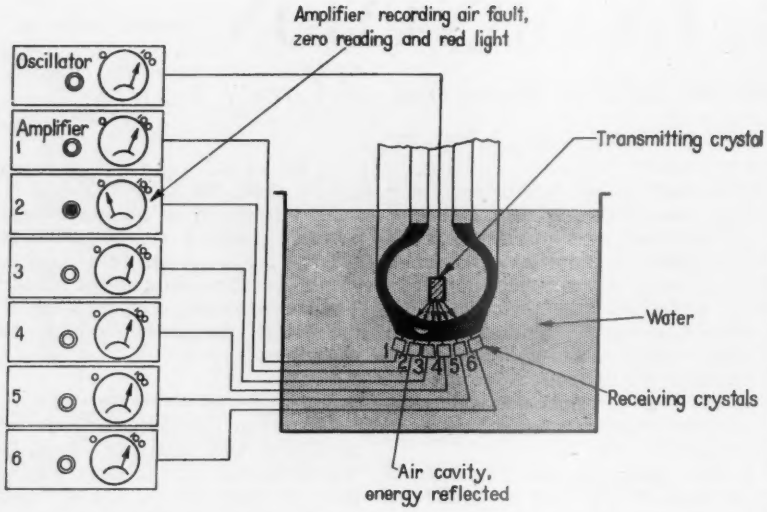


Fig. 2. Diagram of ultrasonic apparatus for detection of air cavities in tyre covers.

crystal electrodes. One way of doing this is to stick a rubber pad onto the front face of the crystal, which is then immersed in water. The rear face of the crystal has an air backing and energy is not radiated from this surface but is reflected and eventually radiated from the front surface.

Fault detection

The early methods for detecting internal air cavities were developed just before the Second World War. The methods were designed to detect small cavities less than $\frac{1}{8}$ in$\times\frac{1}{8}$ in area in metals. A typical method is shown in Fig. 1. Continuous ultrasonic waves are emitted from a quartz crystal into a liquid and pass through the metal under test. These are then picked up by the receiving quartz crystal. This crystal converts the ultrasonic energy into electrical energy which is amplified and then rectified to give a meter reading. The amplifier is of the type used in radio sets except that the meter replaces the loudspeaker. If there is air present in the solid, the ultrasonic energy is reflected from the air layer, no energy is picked up by the receiving crystal, and the meter reads zero.

Unfortunately there is a serious drawback to this system. The ultrasonic waves are sent out in a parallel beam and quite large reflections of the waves occur at the interface between the water and metal under test and also between the water and the receiving crystal. These reflected waves interact with the transmitted waves, and a stationary wave system is set up, as shown in Fig. 1. The signal amplitude at positions a, b, c, which are half a wavelength apart is zero, and if the receiving crystal is placed at a, b, or c, it picks up no signal and hence gives a false fault indication. At 1 Mc/s frequency the distance apart of a, b, and c, in water is $\frac{3}{4}$ mm, so they are fairly close together and any slight alteration in the system could easily give spurious results. To overcome this defect, ultrasonic pulse systems have been developed here and in America, which are similar to radar systems. In this system a transmitter sends out a pulse of high frequency ultrasonic energy and the time taken for an echo to come back is measured. An air cavity will give an ultrasonic echo.

This method, which is used in the metal industry, is unfortunately not practicable for testing tyres because a pulse system requires high-frequency ultrasonic waves which are rapidly absorbed in rubber. By using low-frequency ultrasonic systems, stationary wave effects are avoided, and such a system is used for examining tyres. However, it should be remembered that the main reason for using high-frequency ultrasonic waves for detecting air cavities, is that the higher the frequency the smaller the area of cavity which can be detected. For a cavity $\frac{1}{8}$ in$\times\frac{1}{8}$ in area, an ultrasonic wave at 50 kc/s frequency, that is a wavelength of 3 cm in rubber, would completely pass round the fault and the fault would not be detected by a transmission method as shown in Fig. 1. Fortunately faults which need to be detected in tyres are usually greater than $\frac{1}{8}$ in$\times\frac{1}{8}$ in, so that it is possible to use the lower ultrasonic frequencies. At 50 kc/s frequency the transmitted beam has a semi-angle of about 60 deg for a 1 in diameter quartz crystal. Such a system at this frequency can detect faults $\frac{1}{4}$ in$\times\frac{1}{4}$ in area but is more successful for finding faults greater than $\frac{3}{8}$ in$\times\frac{3}{8}$ in.

Two further advantages of using a low frequency may be noted. First, because of the wide angle of radiation, stationary waves are not formed. Although the transmitted and reflected rays do interact, the effect is a minor one. Secondly, the wide angle of radiation makes it possible to use six receiving crystals with one transmitting crystal. A similar method (B.P. 571817) using a much lower frequency still, has been described in America, but the details given differ radically from the method described here. The basic method we have adopted is indicated in Fig. 2. which shows a cross-section of a tyre in a bath of water. Ultrasonic waves at about 50 kc/s frequency are generated by applying a varying electric field to the transmitting quartz crystal, which is mounted in the well of the

Fig. 3. Prototype equipment for tyre tests.

tyre. The inside of the tyre is filled with water. The transmitting crystal sends out a side beam of ultrasonic energy into the water. This energy passes into the tyre without reflection at the water-rubber interface. The absorption at these frequencies is relatively low. The waves pass through the tyre into water on the far side of the tyre and are detected by six identical X-cut steel-quartz receiving crystals. These crystals convert the ultrasonic energy into small amplitude electrical waves of the same frequency, which are electrically amplified and are then observed as a meter reading.

A certain amount of scatter of the waves occurs in the tyre because the textile cords are full of air. The difference in rubber thickness across the tyre section is not important so long as the tread pattern is free of air bubbles. At a higher frequency, high absorption in the rubber would make tread pattern irregularities a disadvantage in testing. If any air cavity is present in the tyre, such as loose tread, loose plies or cord break-up, then the waves are reflected at this interface and no ultrasonic energy is picked up by a receiving crystal in line with this fault. A zero reading is therefore indicated and this lack of signal, in addition to giving a zero reading, causes a red light to operate. This is shown diagrammatically on channel 2 in Fig. 2.

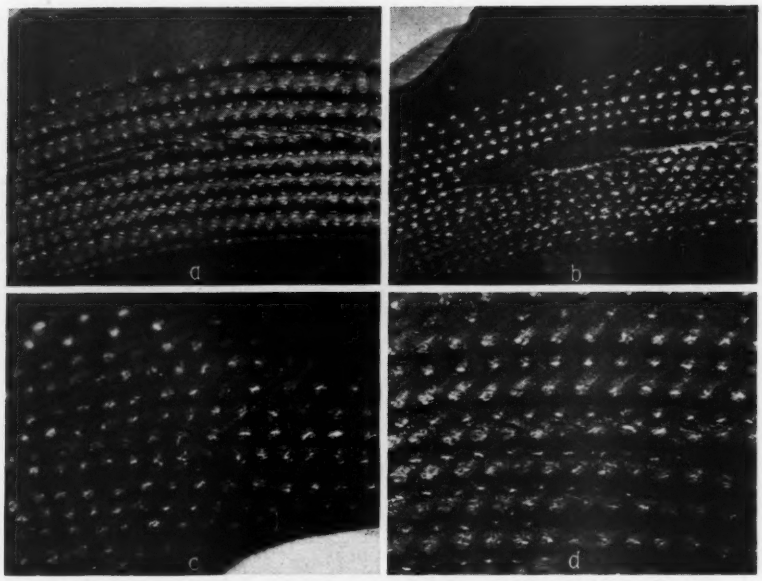


Fig. 4. Air cavities detected by ultrasonic apparatus. Signal drop (per cent) a—100, b—100, c—30, d—50.

Prototype apparatus

The complete prototype apparatus is shown in Fig. 3. This has been extensively used for inspecting a wide range of tyres. On the left of this photograph is the electrical apparatus. The uppermost unit is a rack which contains the oscillator and below this are mounted six separate amplifiers. The leads from the oscillator to the transmitting crystal and from the receiving crystals to the amplifier are taken by co-axial cables into and out of the water tank shown on the right. Control "A" on the amplifier varies its gain. The meter reading can be varied and is usually set at 80 per cent full-scale reading. Control "B" adjusts the amplifier reading at which the red light "C" operates. It is usually set at 20 per cent full-scale reading. It therefore follows that a drop in signal strength from 80 per cent to 20 per cent operates the red light. For practical purposes it is not advisable to interpret readings of signal strength with a reduction of less than 40 per cent.

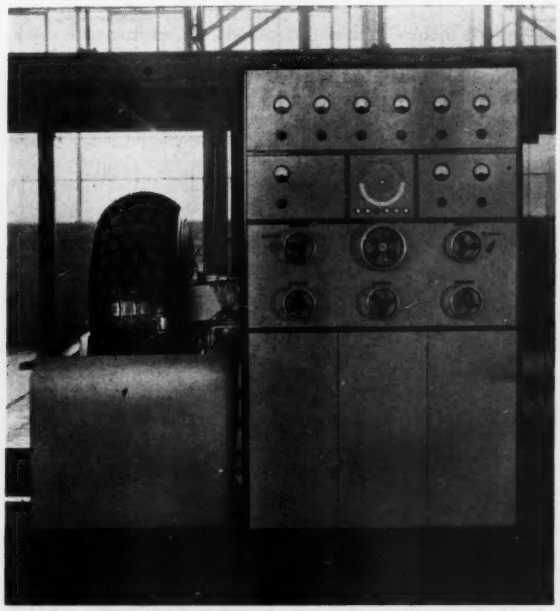


Fig. 5. Modified equipment arranged for routine testing.

The mechanical apparatus consists of a large water tank 5 ft × 2 ft × 2 ft 6 in and a robust iron framework which carries two overhanging 12 in × 1½ in. diameter detachable rollers. The end of each roller is tapered and fits into a socket so that it can be removed for positioning a tyre in the water tank. The rollers are driven through gears by a ⅛ h.p. motor which can be controlled from 0 to 50 r.p.m. It has been found that this motor will easily rotate tyres weighing up to 400 lb. Each roller is provided with stops to keep the tyre in position and prevent it rolling off sideways. The water level usually covers the bead of a tyre at its lowest point in the tank.

The receiving crystals are mounted on a brass channel bent to a suitable curve; on this channel runs a brass trolley which carries the six receiving crystals positioned around the outside section of the tyre. The transmitting crystal is placed in the well of the tyre and the direction of the ultrasonic beam can be altered to radiate through the crown or sidewalls of the tyre. The transmitting crystal usually points directly at the middle receiving crystal.

Examination technique

The tyre is placed on the two rollers and is at first rotated as quickly as possible. This usually removes surface air bubbles, but the wetting of the tyre can be improved by using a small amount of carbowax in the water. Detergents like Teepol X can be used to wet the tyre, but Teepol tends to foam, although this can be minimised by using an anti-foam agent. Having wetted the tyre, the transmitting crystal is placed inside it, about 1 in from the inner tyre surface, and the receiving crystals are positioned in line with the transmitting crystal.

The readings of the amplifier are adjusted to about 80 per cent full-scale value, and the tyre is then rotated. If one is only looking for appreciable fault areas greater than 1 in × 1 in, then the tyre can be rotated at a maximum speed of about 6 r.p.m. for a giant tyre. To detect smaller fault areas it is advisable to rotate the tyre slowly. In one revolution of the tyre about a 5 in arc of section is scanned. To examine the rest of the tyre the transmitting crystal and the trolley carrying the receiving crystals are moved to scan a fresh arc of section and the tyre is again rotated. In this way the complete tyre can be scanned in a few revolutions.

On rotating the tyre small oscillations of about 20 per cent occur on the meter needles. This is due to variation of tyre structure, such as ply overlaps and stray radiations reflected from the side of the tank. With experience an operator can distinguish between variations caused by a small air fault and spurious variations. Air faults cause a very steady change, the false variations cause rapid fluctuations. For big fault areas greater than 1 in × 1 in, little skill is required to operate the apparatus. The sharp drop of the meter needle provides convincing evidence.

This ultrasonic method locates the fault position, but does not indicate precisely whether the fault occurs in the tread or fabric part of the tyre. The method will detect air cavities in the tread between tread and casing, between the plies, and inside the plies due to cord break up. Fig. 4 shows typical faults which were detected by ultrasonic testing. In this illustration (a) and (b) show very bad ply looseness, both of which gave a signal drop of 100 per cent in the detecting meter; (c) and (d) show the start of incipient ply looseness which gave a drop in signal strength of only 30 per cent and 50 per cent respectively.

Routine testing

It was found that the prototype apparatus, Fig. 3, had a number of mechanical disadvantages, and an improved apparatus has recently been produced. Both the mechanical and the electrical equipment were built by the General Electric Co. Ltd. As shown in Fig. 5, the control gear and the indicators linked to the various crystals are on the right and a spare transmitter and oscillator has been provided, although service interruptions due to electrical faults in the prototype apparatus have so far been negligible. A tyre is shown in position in the tank and the leads to the crystals can be seen entering the tank at its right-hand corner. The receiving crystals are driven round the supporting channel by a motor and the water is removed from the inside of the tyre by a suction pump. Between the tyre and the control unit can be seen the pneumatically operated hoisting arm that raises the tyre clear of the tank, moves it laterally, and lowers it to the ground for replacement.

Acknowledgments

The author wishes to thank the Technical Director, Dunlop Rubber Co., Research Centre, Fort Dunlop, Erdington, Birmingham, for permission to publish this Paper. Thanks are also expressed to Mr. E. F. Powell, Physical Research Manager, Dunlop Rubber Co., and Messrs. C. Cropper and I. Farren of the General Electric Co., Research Laboratories, Wembley, for their very valuable contributions in the development of this work. Finally, thanks are expressed to Messrs. T. Julian and Mr. V. Warner, General Electric Co., Service Department, Westminster, who designed and built the latest apparatus on the basis of the Dunlop prototype.

RUNNING IN

A PAPER published in *Automotive Engine Rebuilders Association Proc.*, May, 1950, entitled "*Relationship of Piston Rings and Cylinder Finish*," by B. A. Yates, states that the practice in the aircraft engine industry is that after the running-in period, the engine is refitted with new piston rings and operates successfully under heavy-load conditions with practically no additional running-in. This shows that running-in is unnecessary for the ring face surface, but that, despite the accuracy of the bore and its surface finish as machined and honed, some additional treatment of the cylinder surface is needed.

Running-in appears to consist of the wearing away of the high spots and distorted areas of the cylinder bore surface, and is accomplished by the piston rings throughout the cylinder bore ring travel length. The amount of metal removed is infinitesimal. Very slight distortion may be caused by changing temperature, and distortion of the cylinder surface may also arise from the imposition of mechanical stress. Since the generation of a satisfactory work surface must take place under changing conditions of temperature and load the best method is actual operation of the engine for a running-in period.

Cylinder bore surfaces with a finish between 15 and 35 micro-inches R.M.S. will, after suitable running, show perhaps 8–12 micro-inches R.M.S., and subsequent performance will be satisfactory. Cylinder bore surfaces finish-honed to the same micro-finish of 8-12 micro-inches R.M.S. perform very poorly when placed into operation and subjected to a running-in period. The 8 micro-inch R.M.S. finish produced by running-in seems to be accompanied by micro-changes of the surface which offset distortions from changing temperatures and mechanical loads.

Piston ring finishes are specified in terms of the number of serrations per inch and the type of tool used. The depth of serration on the face may vary from 200 to 300 micro-inches R.M.S. Surface treatment of the chemical and oxide types serve to protect the mating parts during the running-in period and eventually wear away. The inherent hardness of chrome-plated rings makes for metal removal from cylinder bore rather than from ring face; hard-type plating has recently been successfully used instead of the porous type. Cast-iron cylinder bores and piston rings have good resistance to dry wear, the graphite contained in the material providing a measure of dry lubrication when the oil film breaks down. (*M.I.R.A. Abstract* 5333).

SHOW REVIEW NUMBER

The extra issue of the "Automobile Engineer" will be published as usual in connection with the London Motor Exhibition. It will constitute a critical review of the more interesting exhibits, including coachwork, and will have numerous illustrations of special features and design characteristics.

Publication of this issue will follow the ordinary November issue, which will appear on November 1st. The Show issue will be available on November 15th, and can be obtained by order from newsagents throughout the United Kingdom, price 3s. 6d. net. Readers are reminded that it is still necessary to make arrangements with the newsagent to make certain that a copy is secured.

W. L. Fisher, M.I.Mech. E., M.S.A.E.

As announced on page 357, Mr. W. L. Fisher has retired from the editorship of this journal after nearly 40 years in the editorial chair. His business address is now 75, Victoria Street, London, S.W.1. (Telephone number Abbey 3769.)

NEW PLANT AND TOOLS

Recent Developments in Workshop Equipment

A NEW simplified type of single spindle automatic machine, illustrated in Fig. 1, has been developed by B.S.A. Tools Ltd., Mackadown Lane, Marston Green, Birmingham. Production of this additional type has been undertaken because of the present heavy demand for machine tools and the fact that many components can be produced equally well on a simpler and less expensive machine than the B.S.A. single spindle automatics hitherto available. It must be stressed that this new machine retains the accuracy and speed of the other B.S.A. single spindle automatics. These machines will conserve time and materials and will make available a greater number of six-station automatics for the production of more complicated components.

Fig. 1. B.S.A. No. 48 simplified single spindle automatic.

At present two sizes are available, No. 48 and No. 68, but if a demand arises the range will be increased. The No. 48 has a bar capacity up to $\frac{1}{2}$ in diameter and No. 68 up to $\frac{3}{4}$ in diameter. Basically, the construction of these simplified machines is similar to that of the established B.S.A. single spindle automatic screw machines, but in many respects the design has been altered to suit the purposes for which the machines are intended. For example, the extended feed of bar stock permits the production of parts considerably longer than those usually accommodated on automatics. On the No. 68 machine a feed of 9 in can be used.

A self-contained motor mounted at the rear of the machine base provides the drive through vee ropes to the main gearbox shaft. All the gearbox shafts are carried on ball bearings and are lubricated automatically. A range of 10 forward spindle speeds on the No. 68 and 12 on the No. 48 can be obtained by changing one pair of gears. The spindle speed ranges are 474–4480 r.p.m. on No. 68 machines and 1005–6020 r.p.m. on the No. 48 machines.

To allow high speeds and heavy cuts to be employed, the plain work spindle is rigidly constructed and is mounted on roller bearings. A roller chain is used for the drive from the gearbox to the spindle. An adjustable jockey sprocket is used for maintaining the correct chain tension. Adjustment is made manually on the No. 68 machine. On the No. 48 machine it is effected automatically, since on this smaller machine more frequent adjustment of chain tension is necessary.

A front camshaft controls the cross slides, stock feed, work chute, and collet opening and closing. It is driven by the lead camshaft from the back shaft through a worm drive and pick-off gears which provide different cycle times entirely independent of the spindle speed. On the No. 68 machine, provision is made for 65 cycle times ranging from 2 to 180 seconds, and on the No. 48 machine, for 46 cycle times in the range $\frac{3}{4}$ to 75 seconds. Cycle time changes are effected easily and rapidly through pick-off gears at the end of the machine.

The backshaft is driven at constant speed by a belt from the main drive shaft. It is engaged or disengaged by means of a lever that operates a clutch. A handwheel is provided for manual operation of the machine for setting-up purposes. Dog clutches to control the stock feed, and collet opening and closing cams, are mounted on the backshaft and are operated by adjustable trips mounted on the front camshaft. The cross slides are operated by disc type cams mounted on the front camshaft. They are totally enclosed at the front to exclude swarf. A third, or vertical, slide can be fitted when required. The motion for the third

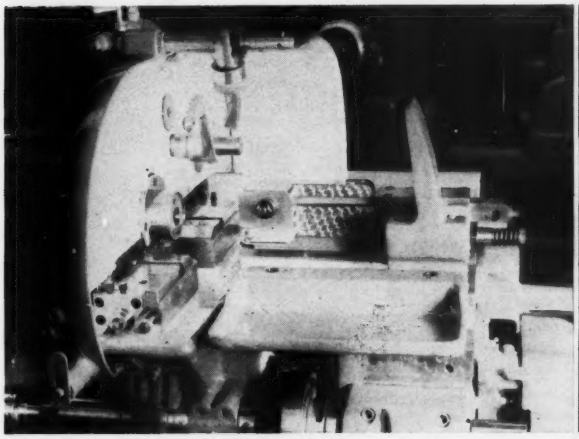


Fig. 2. Tool slide on the B.S.A. simplified single spindle automatic lathe.

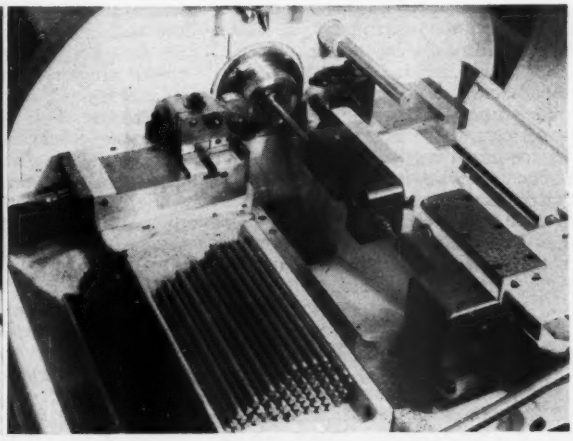


Fig. 3. Combined tailstock and steady in place of a third tool in the tailstock slide.

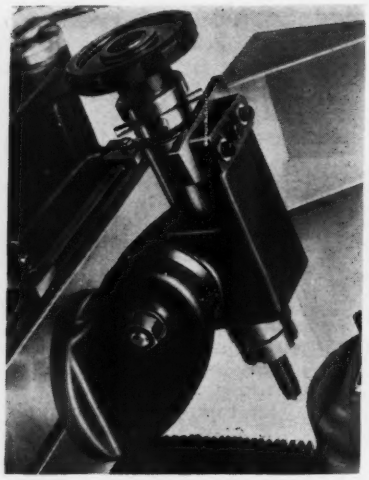


Fig. 4. Herbert Tapconer for grinding the taper lead of taps with relief on the lands.

slide is taken from an additional cam on the front camshaft. Each slide can be set independently either forward or backward in relation to the centre line of the spindle. The tool slide is shown in Fig. 2.

On these simplified automatics, the conventional indexing turret is replaced by a plain slide that carries one tool holder which will accommodate any turret tool of standard type. When the component is such that only two tools are necessary, they will be mounted on the cross slides, and the tailstock slide can be used as a simple bar stop. Alternatively, if the components are long, a combined revolving steady and stop can be fitted, see Fig. 3. On the No. 68 machine the tailstock slide moves on large square section ways, while on the No. 48 machine dovetail ways are used. Each type incorporates a taper gib for taking up wear. Feed is actuated by a cam-operated rack and quadrant. The separation of the work and swarf is effected by the work chute which is operated from an adjustable dog on the camshaft.

There is efficient guarding to protect the operator, and various devices to protect the machine have been embodied in the design. For example, should the machine be subjected to undue strain, a slipping clutch in the cycle time gear train comes into operation and instantly stops all moving parts other than the spindle. Instantaneous stopping is also effected when the remaining bar stock is insufficient to complete another component. For this, the actuating device is incorporated in the bar feed mechanism.

Herbert Tapconer

A simple but useful attachment developed by Alfred Herbert Ltd., Coventry, for grinding the taper lead of taps with relief on the lands is illustrated in Fig. 4. It is intended for use on the Herbert junior surface grinder or other similar machine. The base plate, which can be bolted to the machine table, carries a bracket that swivels in the vertical plane. Graduations numbered 1, 2 and 3, are provided for setting the bracket to required angle for first, second and bottoming taps respectively.

A spring-loaded spindle fitted with a split collet for gripping the tap is housed in the bracket. Provision is made for both rotary and linear motions of the spindle. The combined movement is controlled by a fluted knob, and adjustable straight line cam and a collar with three or four projecting pins according to the number of flutes on the tap. The amount of relief is determined by the setting of the cam plate which swivels on a central pivot and is clamped by two screws. Four index marks on the top left-hand surface of the cam plate are provided for setting the correct relief. Two are for right hand taps and two for left hand.

To set up the attachment the tap is inserted in the collet, and one pin of the collar at the rear end of the spindle is pressed against the underside of the cam plate. The tap is then rotated in the collet until one cutting edge and the opposite heel edge are in the horizontal plane. When this setting is effected the collet is closed. The right hand table stop is then set to bring the tap directly under the grinding wheel. The table is then raised to put on the required cut and the fluted knob is pulled forward and, for a right hand tap, rotated clockwise allowing the pin to ride down the face of the cam until the next pin touches the underside of the cam. This process is repeated until all the lands have been ground. Spiral taps can be ground by setting the attachment so that the axis of the wheel spindle is approximately in line with the spiral angle of the tap. If the chamfer of the tap is longer than the width of the wheel the attachment can be set so that the spiral angle is in line with the length of the table. A small diameter wheel is desirable to give maximum clearance.

Radial drilling and boring machine

A 5 ft radial drilling machine designed and built by the Ajax Machine Tool Co. Ltd., Halifax, is illustrated in Fig. 5. A new factory has been equipped for the quantity manufacture of this machine which is designed to allow the maximum advantage to be taken of the metal removing capacity of modern cutting materials. The pillar is fitted inside the sleeve. This gives extreme rigidity and permits locking arrangements superior to those on machines without an internal pillar. Instead of only relatively small area locking pads, the large areas allow the saddle and arm to be securely locked against the heaviest cutting conditions. No extra lock is required under any conditions.

Supporting the arm is a heavy 13 in diameter cylindrical section cast iron sleeve. The upper end of the sleeve is suspended on the internal pillar and the thrust is taken by a heavy ball thrust bearing. The lower portion of the sleeve is of larger diameter where it is fitted to the pillar base. A combined ball and roller bearing takes the pressure due to the weight of the arm, and the extended portion of the sleeve at the base carries locking gear which secures the sleeve to the pillar. In effect, this construction makes the sleeve and the pillar a single column of great strength.

The high grade cast iron arm swings in a circle. It is a well ribbed box section of ample width and depth to prevent distortion in all working positions. A patented design switch gear is housed inside the arm casting. The main casting for the saddle is totally enclosed to give the best possible working conditions for the gearing and mechanism. A vee slide on the lower portion of the saddle is bedded

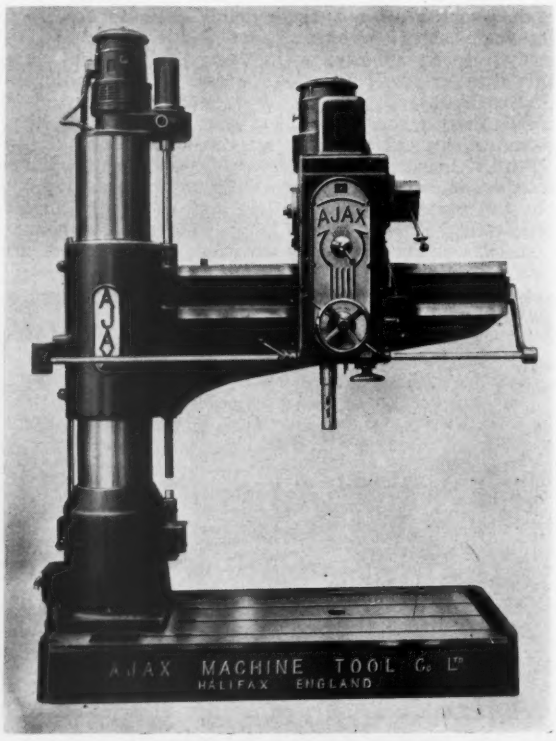


Fig. 5. Ajax 5 ft. radial drilling machine.

to the radial arm to give true spindle alignment. Rollers, with ball bearing spindles that travel on machined surfaces within the front of the arm, take the dead weight of the saddle. Rapid adjustment of the saddle along the arm is through steel rack and pinion actuated by a conveniently placed handwheel. All the gears in the saddle are made from heat-treated nickel-chrome-molybdenum steel.

Combined locking is effected by movement of a single lever which simultaneously locks the saddle to the arm and the sleeve to the pillar. The arm cannot swing in the locked position. Therefore it can be elevated in the locked position while still retaining the true drilling position. Large bearing surfaces in the arm casting bores permit rigid locking to withstand heavy drilling conditions. The saddle is positively locked by drawing the vee slide against the corresponding bearing on the arm. This ensures true vertical alignment of the spindle.

An electric motor mounted directly on the saddle provides the drive for all spindle speeds and feeds. An auxiliary motor on the top of the column elevates and lowers the arm. Both motors are specially constructed to withstand conditions of repeated reversals. The spindle is of high tensile steel. It is accurately ground and is mounted in hard phosphor-bronze bearings. Ball thrust bearings are fitted to take the thrust in both upward and downward directions.

There is an automatic trip motion to prevent over-feeding of the spindle in either direction. It can be adjusted for drilling to predetermined depths. The spindle has 14 in of feed and is supported through its entire travel by a steel sleeve which has a rack cut in its outer surface to engage with a steel pinion that transmits both power feed and rapid adjustment. A special slow feed for fine adjustment is also provided. Spindle balance is effected by a compensating spring device which operates in all positions of travel.

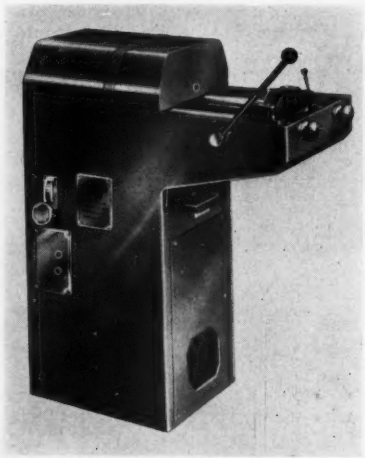


Fig. 7. Staffa single spindle high speed screwing machine.

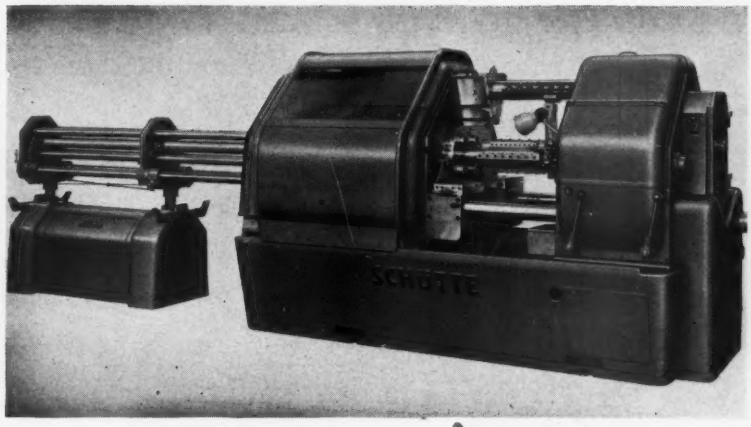


Fig. 6. Four-spindle, 1⅜ in. capacity Schütte automatic lathe.

There are eight spindle speeds in a choice of three ranges. Changes are made by means of a single lever that operates through a gate. The selected speed is securely retained by means of a plunger stop. This eliminates internal spring keys. The speed in r.p.m. is clearly shown at each station. Four feed rates are available at each of the eight speeds. The feed worm is of hardened steel and is engaged with a helical gear. Simple motion of levers in the front of the saddle gives instantaneous engagement of all 83 teeth in the clutch. Start, stop, reverse and inch motions are controlled through a specially designed switch on the saddle. The number of possible reversals per minute is limited only by the skill of the operator and the cutting speed of the taps.

Multi-spindle automatics

An entirely new four-spindle, 1⅜ in capacity automatic lathe was exhibited at the recent Hanover Fair by Alfred H. Schütte. It is shown in Fig. 6. Some unusual features are embodied in the design. Two camshafts are provided. They run along the side of the machine. The cam levers are adjustable to allow one set of cams to cover a wide range of feeds. Another important feature is that the tools are easily accessible. This simplifies tool setting and also ensures easy swarf removal.

In contradistinction to the arrangement on conventional four-spindle automatics where there is only one longitudinal slide with four tool holder positions, this machine has four independently-operated longitudinal slides. It is therefore, possible to select the most appropriate feed for each spindle position. The four independently-operated cross slides can be augmented by fitting a fifth and sixth cross slide.

This machine is also fitted with an automatic spindle stopping mechanism that comes into operation when the last piece of bar is too short to produce a component. This eliminates the danger of tool breakage when such conditions arise. The bar guides are of the retractable, sound-resisting type. All the usual attachments, such as tapping, screwing, chasing and high-speed drilling, are available for use with this machine. Rockwell Machine Tool Co. Ltd., Welsh Harp, Edgware Road, London, N.W.2, are the agents for these machines.

Single spindle screwing machine

An addition to the range of products manufactured by Chamberlain Industries Limited, Staffa Road, Leyton, London, E.10, is illustrated in Fig. 7. It is a single spindle screwing machine for the accurate high speed threading of bolts and small components. The pedestal of the machine is of fabricated construction of a type that gives the rigidity necessary for accurate high speed threading. A hollow spindle is employed. It is designed to take a rotating diehead and runs in self-lubricating Oilite bearings with thrust washers of the same material to take end loads during the cutting operation.

A small gear pump, housed within the reservoir, pumps the cutting lubricant. Any leaks from the pump are trapped in the reservoir so that there is no danger of contaminating the driving belts and pulleys and so causing belt slip. The cutting lubricant is delivered through the hollow spindle on to the work to wash swarf away from the die-head and to keep both sides of the chasers clean. To seal any oil leaks at the end of the spindle, an oil retainer is incorporated in the rear bearing housing. The spindle and lubricant flow control fittings are totally enclosed by a removable push-on cover. The diehead is also enclosed by a hinged cover.

A flat top table to which any form of self-centring vice or work-holding fixture may be mounted, is incorporated in the design. Oilite bushes are fitted to minimize the frictional wear caused by the sliding movement of the carriage. Two tubular guide bars, running the length of the cradle, provide the slide bed. They are extended on either side of the front bearing housing. These guide bars maintain the correct alignment of the table top horizontally and

vertically with the centre of rotation of the machine spindle. The action of a lever swinging through an arc of 90 deg and coupled to a single linkage mechanism at the rear end of the cradle, controls the forward and return movement of the carriage assembly.

For setting the length of thread to be cut, two screwed shafts with trip adjusting nuts are housed in the tubular slide bars. A crosshead bracket positioned inside the machine pedestal couples two ends of the screwed shafts together. From this bracket the yoke of the diehead is automatically controlled. The trip adjusting nuts are positioned on either side of the carriage table, and are adjusted inside the slide bars by means of two control knobs at the front end of the cradle. These knobs are graduated to indicate the length of table movement. As the carriage does not incorporate springs or return weights, there is no drag on the work that might cause deformation of the thread. The forward and return action of the carriage through register against the trip nuts, gives automatic opening and closing of the diehead. As a safety precaution, the diehead can be opened by control knobs at the front end of the table.

Drive to the spindle is by a single vee belt running on an adjustable cone pulley. The variable drive assembly is mounted on a quadrant shaped bearing, and the range of speeds for threading various materials is obtained by adjustment of the quadrant. A control handle on the side of the machine operates the quadrant and the speed at which the diehead is running is shown on a graduated scale attached to the quadrant. The speed range is from 266 to 1,007 r.p.m. and the machine has a capacity of from $\frac{3}{16}$ in to $\frac{3}{8}$ in Whitworth and from $\frac{3}{16}$ in to $\frac{1}{2}$ in B.5. fine.

ACCESSIBILITY

Design in Relation to Maintenance on Buses in the United States

A PAPER entitled " Accessibility of Buses for Maintenance," by R. Whitfield, published in the *S.A.E. Quarterly Transactions,* July 1951, deals with an aspect of design that, while given some consideration, evidently receives less attention than it merits. Thirty companies, operating buses of all makes, contributed to the Paper.

In the United States, labour charges for the repair and maintenance of buses are greater than the fuel costs and account for fully half the total maintenance bill. Improved accessibility directly reduces this expenditure. Having a good effect upon the standards of maintenance, improved accessibility leads to a reduction in the number of replacement parts consumed and so, indirectly, to further labour saving. The buses themselves spend a smaller proportion of their time in idleness.

Compromise between accessibility and other desirable qualities is unavoidable. The purchaser often specifies certain auxiliary equipment by make, increasing the designer's difficulties. The wealth of equipment carried by American passenger vehicles, including automatic and safety devices, provision for heating and ventilation, indicators, lighting and compressed air systems, add complexity to a problem the solution of which in a plain truck would be difficult enough. To attain maximum payload per unit length necessitates the location of engines in odd places at the rear or side of, or underneath, the vehicle, and remote-control devices are required. Manufacturers may forget that components such as carburettors, distributors, starters, clutches and transmissions, installed by the assemblers in convenient sequence before the engine is in place, have to be removed by the maintenance mechanic without d i s t u r b i n g the engine.

Priority in the matter of accessibility has to be given to those items serviced most frequently, and these include dipsticks, in some instances so located that replacement at night requires a flashlight or periscope. Fillers for fuel, oil, transmission fluid, and water should be grouped together and their caps should be hinged and self-closing. Hand catches are preferred to keys for large inspection doors, and air-tank drains should be operated by a single handle. Cleaning is simplified by the use of seats supported only at the wall and on a single pedestal, by attaching stanchions to seat frames rather than to the floor, by provision of generous radii at the corners of floors and steps, and by reduction of offsets or projections so that external side areas become accessible to the brushes of automatic washing machines.

Second in order of importance are the inspection and minor repairs associated with preventive maintenance. On one or other of the various makes of bus, almost every component was reported to be insufficiently accessible. In particular, more attention could be paid to accessibility of sparking plugs, distributor points, oil-bath air cleaners, windscreen wipers, fuse panels, lubrication fittings, and carburettors with their connecting linkages. The adjustment or removal of one component sometimes involves the removal of another component. A manifold may prevent the adjustment of cylinder head bolts, or a shaft may have to be removed before a generator belt can be replaced. In one example, two-dozen panel screws barred access to a door mechanism; in another, an oil-pump had to be taken out before the sump strainer could be removed. Complaints that timing markings on flywheels lacked clarity, for example, were general. Lack of identification of fuses and wires, and the makers' failure to follow their own wiring diagrams were a major source of trouble.

Removal and replacement of complete units by relatively unskilled labour, specialists properly equipped being employed for the repair function itself, minimizes a vehicle's period in shops. Sometimes replacement is arranged to take place after a prescribed mileage. In any event, this system stresses a third set of requirements. There must be space enough for those wrench and spanner manipulations associated with removal and replacement. Care has to be taken that pipes, wires and other parts do not foul the units. Simple means must be sought of ensuring, by indexing or otherwise, correct positions of flywheels on crankshafts, ignition distributors, and other units or components implicated under the term " timing." These components should be disturbed as little as possible when the units are changed, because a skilled mechanic is needed for their adjustment. Positive and negative terminals of batteries should differ in size, to obviate damage to batteries and generators, and identification of wires by tabs, for example, saves time. Units difficult to remove included engines, radiators, heavy generators (in one bus, the air compressor had first to be removed), and storage batteries. The split-type rear-axle housing, as compared with the banjo design, increased the labour involved in changing the differential unit.

Of eight bus-builders consulted, all employed procedures aimed at eliminating maintenance snags, which nevertheless arose. It is difficult to foresee everything encountered in service, even with the help of models. Co-operation of field service engineers with designers helps to check repetition of mistakes, but new mistakes occur. Designers need to have had maintenance experience for some years, in which they worked with the mechanics, to learn, among other things, that exhaust manifolds are hot. Standardizing the connections for hydraulic fluid, fuel, air and oil pipes, and the design and location of towing-gear, would also help appreciably, as would a reduction in the number of spanners now required for an assortment of ordinary and special screw heads. If new screw heads are superior, the best form should be used throughout the bus. Maintenance men can help by drawing the attention of their own management, and of the manufacturer, to unsuspected defects, and by stressing the superior value of buses having good accessibility. (1983)